FINAL REPORT

ME 4182 DESIGN

"Design of a Lunar Backhoe"

Submitted to Mr. Brazell

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ABSTRACT

This report examines the the design of a backhoe to be used on the moon's surface. Several modifications had to be made to the existing conventional backhoe to make the backhoe suitable for the lunar environment. Among the problems encountered include dissipating heat, controlling the backhoe in a soundless vaccuum, and accounting for the difference between the earth's and the moon's gravitational fields. This group was left free to determine its own design parameters for backhoe performance.

PROBLEM STATEMENT

Background

NASA is currently looking into the feasibility of building a lunar base station. Out of necessity, there is now a requirement that specialized earth moving equipment be designed. This equipment must be able to work in the lunar environment as well as have the capacity to perform its function on the lunar soil. This particular team has been designated to design a lunar backhoe. The design will incorporate some of the basic kinematic and fundamental principles of an analogous earth backhoe. However, there will be modifications to the controlling (maneuvering) system and power system of the backhoe. This team has been left free to set its own design and performance specifications.

Performance

The backhoe must be able to interface with the given tractor design. This interface will consist of both a physical coupling and, in this case, a hydraulic coupling. The hydraulic lines will be shielded to prevent high heat buildup due to the constant solar radiation. As a further precaution, the hydraulic fluid will have a flash point in excess of 400 degrees F (205 degrees C). The backhoe will exert high digging forces, which, along with the weight of the backhoe, will have to be counterbalanced by the weight of the tractor and outriggers on the backhoe. This digging force, as yet unspecified, will be in the range of 1000 lbs. as a result of the combined cylinder forces in excess of 9000 lbs. The capacity of the bucket will be in the range of 7 cu. ft. and the maximum digging depth will be in the range of 120 in. (304.8 cm). The bucket may be equipped with quick disconnects to allow for the easy installation of other buckets, other tools, etc. The different links will be designed to be lightweight, have low heat capacity, and have high strength to prevent bending. This criterion will give rise to a tube design. The backhoe will also have to be a free standing unit when not in use.

Environmental limitations will necessitate designing a remote control package for the backhoe. These controls may have a resistant feed back loop so that the operator can "feel" the backhoe working, or a relief valve may be placed in the hydraulic system. Both of these precautions will prevent the operator from overloading the system.

Constraints

The main constraint of the backhoe is the lunar environment. Basically, it is in a vacuum, constantly night or day, with a constant influx of solar radiation and one sixth the gravity of earth. This poses several problems for the backhoe. There will be a large heat buildup in any mass put there and the only means of heat transfer will be by radiation. Thus, it will be difficult to dissipate heat due to friction or heat buildup in the hydraulic lines. Since there is only one sixth the gravity as that on earth, it will be difficult to counterbalance the digging forces. Due to the vacuum, it will be necessary for the operator to wear a spacesuit and this will limit the operator's maneuverability and his field of vision. The vacuum will also make it impossible to hear the equipment operating.

The lunar soil is also a limiting factor. The soil is a cohesionless, fragmental rock type. It will be very abrasive to the materials and will necessitate the use of very hard materials. It is also a powdery top soil which will further limit the operators field of vision.

CONTROLS

The open loop control system used to control the backhoe cylinders consists of a stack of six electro-machanical servo-valves, a handheld controller/transmitter, and a radio receiver.

The valves are four-position, two way, closed center directional servo-valves that uses a torque motor to control a spool that diverts the hydraulic fluid into and out of the cylinders. The position of the spool is proportional to the magnitude of current to the motor. The amount of flow of the hydraulic fluid is controlled by the position of the spool. At the null position, no flow passes to the cylinders, and at full rated current to the motor, full rated flow is driven to the cylinders. Six valves are used to control the backhoe: one each for the shovel, bucket, dipstick, swing, and left and right outrigger cylinders. One valve is used to control the two swing cylinders.

The valves that were selected are stackable, with one connection needed for inlet pressure and one for oulet. Full pressure (2500 psi) is available to each cyclinder. To prevent the operator from overloading the system, relief valves are located at the inlet manifold and at each valve stage. Each of these relief valves are to be set at 2500 psi. To limit the maxium force exerted by the cylinders, pressure control valves are to be placed between the valves and the cyclinders.

The maximum temperature under which the valves can be operated is 100 degrees Celsius using Viton seals. With the hydraulic fluid heat exchanger, the valves should be suitable for operating for an indefinite period. Other specific information for the valves is located in the appendix.

The valves are controlled by a radio receiving unit that drives the torque motor with a current proportional to the strength of the input radio signal. The input signal is multiplexed to allow for maximum signal reliability. Other features of the receiver include: environmentally sealed NEMA-4 Junction Box, internal mountings that permit the unit to be directly mounted to the backhoe, a plug in card that mates the receiver to one sending unit so that several backhoes may be operated in one area, and ramp output for the proportional controls for smooth cylinder motion. The receiver requires a 12 volt power source, so that the box can be directly connected to the tractor's battery.

The receiver should be mounted on the tractor where it will not be physically abused and where it can provide clear line of sight to the operator location. It is suggested that the receiver be mounted at the rear of tractor near the backhoe interface. The receiver can be operated at or below 140 degrees Farenheit. Given that the reciever will be stored at 70 degrees Farenheit and that the maximum temperature of the backhoe will be 250 degrees Farenheit, a properly shielded and insulated receiving box could be operated for at least 4 hours under direct radiation. Material suggested for shielding and insulating the receiver are the same as those used to protect the hydraulic lines.

The controlling/transmitting device is a portable handheld sending unit with 6 spring loaded levers for proportional control and 8 spring loaded toggle switches for on/off control. Only the proportional controls will be used to control the backhoe. The sending unit can either be clipped to the space suit of the operator or hung around his neck with a neck strap. The controls can be operated with the operator wearing gloves. The sending unit uses a 12-volt rechargeable battery that provides 4 hours of continous operation. Other features of the sending unit include: environmentally sealed UHF transmitter, custom label for controls, 2 dead man switches (one of whick must be depressed to operate the sending unit), and 1/4 mile line of sight control.

A suggested configuration for the control on the sending unit is provided in Figure 1 . With the controls set up as shown, the operator can use the backhoe for digging by squeezing the dipstick and boom levers together with his left hand. Also, to dump a load of soil, the swing and bucket levers can be squeezed together with the right hand. This configuration will provide maximum convenience for the operator, and was suggested by an actual backhoe operator.

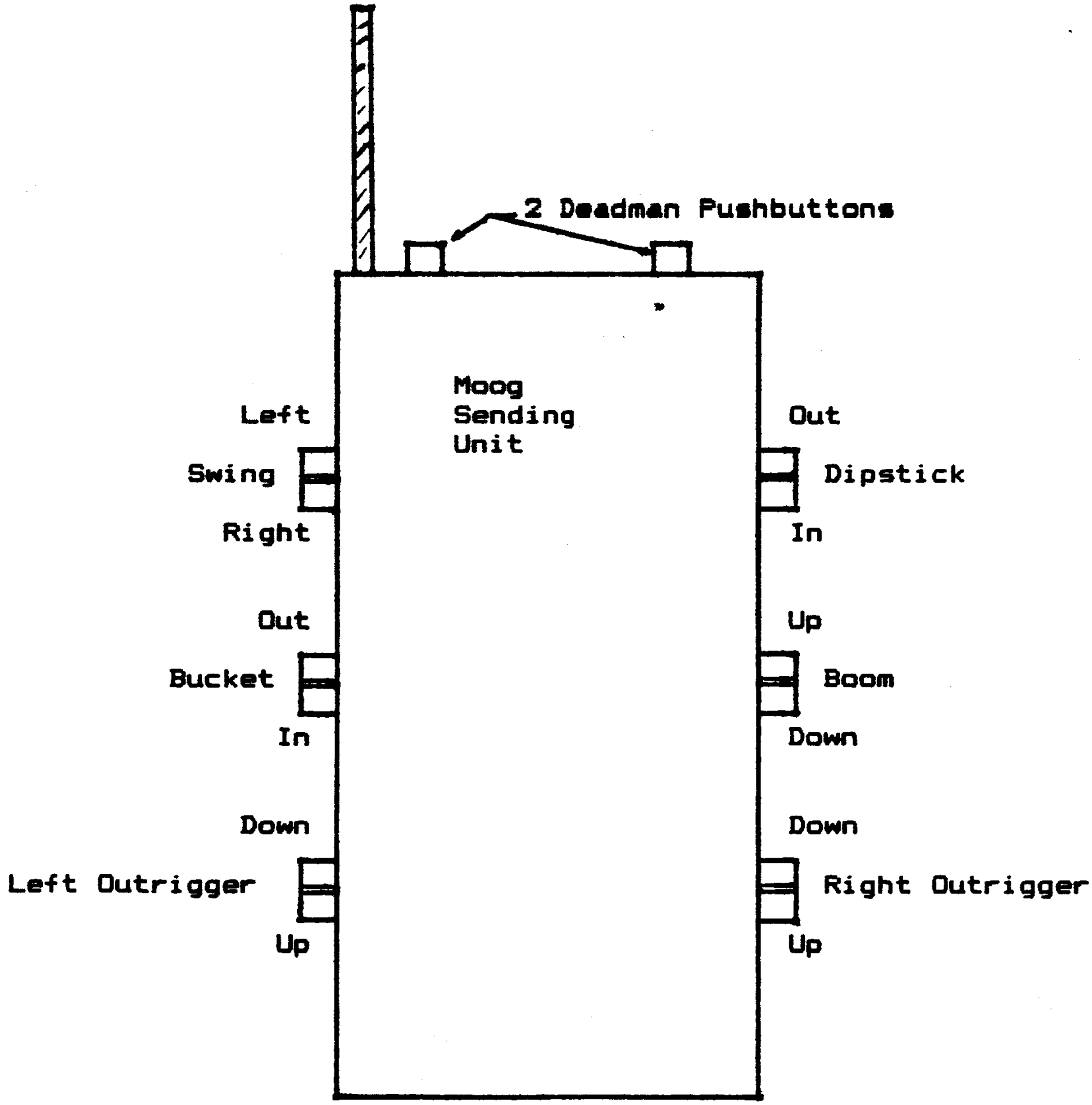
The sending unit converts data from the on/off switches and potentiometers into digitally coded 1's and 0's. Five bits provide for thirty-two possible levels of control where fifteen are for + and - control each and the remaining two levels provide for "control off". Scanning of the switch and potentiometer positions occurs at a rate of twenty times per second. The transmitter is capable of providing four different FM frequencies.

The receiver operates at the same frequencies as the transmitter. An included scanner samples the carrier frequency continuously for a .15 second period. When data is received it is sent to a "shift register" where it is compared to a code card to check for agreement between the code card and sending code. When the comparisons are checked and confirmed, the data is transferred from the shift register to the a position in memory. The data is then processed through buffers and relays for the on/off switches and digital/analog converters for the proportional controls. The next digital data is similarly scanned and if the comparisons are successful, the data enters memory one twentieth of a second later. Included in the receiver are several fail-safe logics. The sending code must be verified for each data set, several successive ON commands must be processed to turn the function on, and "valid transmission" output will return to zero within one-half of a second if succeeding command data is not received.

The control system for the lunar backhoe was designed to be easy and safe to use, easy to set up and service, and with features to prevent the operator from damaging the system. Radio control was selected over manual control to give the operator manueverability to compensate for his inherent difficulties with seeing out of a space suit. Moog was chosen as a vendor for the radio controls because they were the only company that provided a radio transmitter and receiver that could be used directly

Figure 1

Suggested Control Arrangement for Radio Sending Unit



with servo-valves without additional electronic hardware. Their controller is by no means the most convenient one that could be designed for the backhoe operator: a two joystick tummy pack would provide for a system closer to the one on conventional backhoes. The joysticks would also be easier to use with gloves on the operator's hands. But designing such a unique system would introduce complexities that woud require matching many electrical components compatible with the chosen servo-valves. The sender/receiver from Moog was specifically designed to be compatible and electrically matched for the servo-valves. So with this system, we have a tried and proven system with no compatibility problem.

The chosen valves are stackable into a very tight configuration for easy mounting on the backhoe. With the relief valves on the main manifold and for the individual cylinders, the operator or the load cannot easily overload the hydraulic system. Also, with the pressure controlling valves between the cylinders and the valves, the maximum force exerted by the cylinders can be limited. This will prevent the operator from lifting the tractor off the ground.

The specification sheets that were provide by Moog for the valves, transmitter, and receiver are provided in the appendix.

HYDRAULIC SYSTEM - GENERAL SPECIFICATIONS

The hydraulic system is designed to operate using General Electric Versilube F-50 hydraulic fluid over a temperature range 0°F to 200°F. Hydraulic hoses and cylinders which can operate easily under these conditions have been selected and are specified in their respective sections. In addition, provision has been made for a liquid convection cooling system and shielding against solar radiation so that the considerable amount of heat generated under operating conditions can be dissipated.

All cylinders are double acting and honed for concentricity and finish. The piston rods are high-tensile, die drawn steel, hardened, ground, polished, and chrome plated. Cylinder piston packings are of the chevron type. The rod packings are of the chevron type also, but incorporate a viton "U" seal. The packing gland supports and retains the rod packing, bearing sleeve, rod wiper, and backup ring. Cylinder pivot and anchor points utilize self lubricating bushings. The swing cylinders employ a connecting hose at the rod ends of the cylinders that permit oil flow between them.

The main control valve is a stack-type assembly consisting of six sections containing spring centered, remote controlled spools which direct high pressure pump oil to the individual cylinder circuits. Each circuit contains a spring loaded check valve to check the flow from either cylinder port to the valve pressure passage. Also, each section contains adjustable circuit relief valves to protect against pressure overloading. The main control valve is also equipped with an inlet end cover and an outlet end cover. The inlet end cover contains the inlet port and the system relief valve. The outlet end cover contains the return oil port which returns low pressure oil to the reservoir, and the backpressure relief valve. The backpressure relief valve is a simple pilot operated relief valve that functions when the pressure in the return oil passage is approximately 250 psi greater than the pressure in the sump oil passage.

HYDRAULIC FLUID COOLING

Method One

Since the ambient temperature on the moon is high and the enviornment is a vacuum it is necessary to cool the hydraulic fluid. The friction of the fluid will produce much heat and since there is no air, the heat cannot be transferred to it. The radiation will be blocked by the shield; however, the shield will in turn radiate heat to the hydraulic line.

The simplist solution to this problem is to enclose the whole hydraulic line with a fluid. This fluid will be contained in the outer hose of a coaxial hose scheme where there are two hoses, an inner and an outer. The sections of line that are rigid tubing will then be rigid coaxial tubing.

This coaxial tubing is common and can be purchased from System Components in Atlanta, Georgia. There seems to be no manufacturers of coaxial hose and coaxial hose fittings. The hose can be assembled with the recommended teflon hose inside a common two inch stainless steel hose. The fittings can be machined and assembled (see later drawings).

Backhoe Design

All the materials used in the backhoe are made of an aluminum alloy 6066-T6. This alloy contains the strength, workability, and machinability necessary for the tubular design. The boom member is of a C-section design to provide space for the radiator used to cool the hydraulic system. The dipperstick member is a tubular section since the hydraulic components are not internal to that member. All members have been designed as a constant strength beam. This criterion is defined such that the sectional modulus is equal to or greater than the maximum bending moment at a section divided by the maximum allowable stress in a member. The shape of each member follows the shape of the bending moment curve closely, thus defining the tapered section toward the pin ends. All members are a constant 1/4 inch in thickness.The bucket to be used will be a standard steel, 18 inch bucket whose capacity is 4 cubic feet (heap capacity would be approximately 6 cubic feet).The backhoe will be made with a quick disconnect at the bucket interface thus allowing for a different bucket or a different tool to be used. The guide link and bucket link will also be of aluminum. All pins to be incorporated in the backhoe will be of a carbon steel alloy. At each pin connection there will be a pressed fit sleeve that will absorb the forces and distribute them evenly to the aluminum. In this way, stress concentrations will be greatly reduced in the vicinity of the pin connections.

The hitch, which is the physical coupling to the tractor, consists of two parts: the pivot hitch and the interfacing hitch. The pivot hitch will be of cast aluminum 6066-T6. This is where the backhoe connects indirectly to the tractor. The function of this hitch is to produce the swing angle of the boom. It contains the bore hole about which this hitch pivots and it also contains the single pin connection to which the swing cylinders connect. The interfacing hitch, as the name suggests, is the direct connection to the tractor.The interface consists of a hooked bottom and a pin connection at the top; this will allow for easy disconnection and connection. The hitch consists of a four inch tube stock frame with four 1 1/2 inch aluminum plates welded and supported to the frame. The swing cylinders are connected in the center of the two sets of plates. The stabilizer cylinders and legs are also an integral part of this interfacing hitch. The connection between the pivot hitch and the interfacing hitch will be time consuming and complicated. First the pivot hitch is press fitted with a sleeve and inserted between the aluminum plates on the interfacing hitch. Once the bore holes line up, the sleeve is then line bored and the resulting bore hole is fitted with a pin.

We are assuming that the bearing surfaces used are of a non-lubricating type. This will eliminate the need for any high temperature grease or self lubricating bearings.

Hydraulic Cylinder Specific Vendor Information

Carter Controls, Inc. will custom build hydraulic cylinders to the dimensions listed for their Roundline® welded cylinders. These cylinders are capable of extended operation at 450 degrees F and 3000 psi. The required structural specifications were developed in the section on cylinder sizing. Mounting conditions are shown in the main diagram of the hydraulic system.

Bucket Cylinder Special Requirements

1) Pivot Mount Both Ends, Use Eye RE 2.
2) Equip Cylinder with external tubing such that both connections are at cap end- use 3/4" steel tubing with 1 1/16"-12 SAE 37 degree Flared male fittings, facing cap end.

Dipstick Cylinder Special Requirements

1) Pivot Mount Both Ends, Use Rod Eye RE 3.
2) Equip Cylinder with external tubing such that both connections are at center and facing cap end- use 3/4" steel tubing, same fittings as bucket cylinder.

Boom Cylinder Special Requirements

1) Pivot Mount Rod End, Use Rod Eye RE 3. CLevis mount Cap End-RC 3.
2) Equip Cylinder with external tubing subject to same conditions as Dipstick Cylinder.

Swing Cylinders (2)

1) Pivot Mount Cap End of both, rod end of one; use CLevis; mount RC3, rod end of other use Rod Eye RE 3.
2) No external tubing.

Stabilizer Cylinders (2)

1) Pivot Mount Both Ends, Use Rod Eye RE 3.
2) Equip Cylinder with external tubing such that both connections are at cap end, one cylinder with connections at right angle on left, the other cylinder with connections at right angle on right. Use 3/4" steel tubing, same fittings as bucket cylinder.

PIN CONNECTIONS

All the pin connections will be similar. The description of one will be demonstrational for the rest. The main problem is that there needs to be a high strength steel pin connecting the rod eye of the cylinder to the aluminum member. The aluminum member welded up as shown in the drawings, is much softer than the pin and upon any impact, the aluminum will absorb the energy since there is going to be some clearance between them. The solution is to insert an alloy steel sleeve inside the aluminum member by means of a press fit. Thus the sleeve will absorb some of the impact energy while redistributing the remaining energy over a large area of aluminum.

The aluminum member shown in the drawing will be welded as shown and then line bored to exact tolerances. The pin and steel inserts will be machined 4340 steel drawn at 1000°F having a hardness of 50 to 55 Rockwell "C" scale. The machining operation will use a cubic Boron nitride tool which is capable of machine this very hard steel.

The pin will be held in place by the plate coupler on the end. Grade eight bolts will extend through this coupler plate deep into tapped holes in the aluminum member.

Bearings that will be used in the pin connections will be a low speed, non-lubricating bushing type. The bearings are called Metcar^{tm} high temperature bearings and are capable of a working temperature of 750°F and a maximum temperature of 900°F. They can be purchased from Metalized Carbon Corporation located in ossining, New York.

MATERIALS

In the area of structural materials, there is a need to minimize weight and maximize strength and toughness. It was found that high strength aluminum would be the most suitable. Although some high strength steel alloys have a higher strength per pound, the aluminum was found to be more favorable. Three aluminum alloys were under consideration. First, 7050 T736 has twice the fatigue strength of the others and is commonly used for die forgings in aircraft; however, it is difficult to weld. Secondly, 6066 T651 has a UTS of 57000 PSI and is commonly used as forgings and extrusions for welded structures. It is difficult to weld with gas but can be welded by arc. Thirdly, 6061 T6 has a UTS of 42000 PSI and is commonly used in fabrication because of its ease of fabrication. It has excellent welding characteristics both for gas and arc.

The GXXX series aluminum alloy contains silicon and magnesium while the 7XXX contains 1% to *% zinc with smaller percents of magnexium, copper and chromium. The 7XXX series has by far the best strength characteristics; however, due to its poor welding characteristics the 6066 T651 and 6061 T6 are the alloys that are recommended.

COST ANALYSIS

Description	Vendor Part#	Quantity	Price
Note: all hose is Deutsch Co. Teflon High Pressure with Teflon Core and Steel Braid Cover.			
16" Flexible Hose	84A0100012C016	1	$8.00
20" "	84A0100012C020	1	$10.00
72" "	84A0100012C072	6	$216.00
20" "	84A0100012C020	1	$10.00
24" "	84A0100012C024	1	$12.00
16" "	84A0100012C016	1	$8.00
12" "	84A0100012C012	1	$6.00
24" "	84A0100012C024	2	$24.00
40" "	84A0100012C040	1	$20.00
28" "	84A0100012C028	1	$14.00
36" "	84A0100012C036	2	$36.00
54" "	84A0100012C054	1	$27.00
40" "	84A0100012C040	1	$20.00
36" "	84A0100012C036	2	$36.00
24" "	84A0100012C024	2	$24.00
40" "	84A0100012C040	1	$20.00
80" "	84A0100012C080	2	$80.00
32" "	84A0100012C032	2	$32.00
3/4" Steel Tubing, 48" Standard, Use 1 1/16 12 Thread Female SAE 37deg flared connections, both ends		2	$20.00
Bucket Cylinder		1	$300.00
Dipstick Cylinder		1	$450.00
Boom Cylinder		1	$700.00
Swing Cylinder		2	$300.00
Stabilizer Cylinder		2	$400.00
Aluminum Tubing 6066 T6		900 lbs	$2700.00
Manufacturing of Materials 10 men 12 weeks 40 hrs/week $18/hr		4800 man/hrs	$86400.00
Pins and Sleeves		21	$882.00
Sleeve Insert		21	$441.00
Bearings, high temperature Metallized Corp. Ossing, NY		21	$756.00
Additional Labor Hydraulic Lines, etc. 10 men 8 weeks 40 hrs/week 3200 hrs 18$/hr		3200 man/hrs	$57600.00

Coaxial Hose 350 feet			$1750.00
Bucket, Standard		1	$480.00
Rod Eyes		21	$210.00
Shielding			$1000.00
controls			
Note: all parts are provided by Moog Corp.			
Proportional Electrohydraulic Directional Control Valves			
valves for dipstick, boom and swing cylinders	A65-20-3-W-L-250	3	$750.00
valve for the bucket cylinder	A65-20-3-W-L-250	1	$250.00
Inlet Manifold Assembly	A31094-2-250	1	$100.00
Outlet Manifold	A31102-1V	1	$101.00
Sending Unit		1	$100.00
Proportional Plus On/Off Controller	130-180	1	$100.00
Tranmitter	125-104	1	$200.00
Battery Pack	128-103	3	$200.00
Junction Box		1	$200.00
Prototype Production			
Testing			
2500 man hrs			$45000.00
Remodification			

HAZZARDS

The biggest foreseeable hazzard is the hydraulic fluid. The one specified is able to withstand extremely high temperatures. The high temperatures combined with its high pressure, could cause extreme harm to someone if there was a failure in the hydraulic line.

Another hazzard might be in the area of materials. Six thousand and seven thousand series aluminum anneal at 775°F in about three hours. The beginning annealing temperature would be less than that. It is clear that the temperature of the aluminum, when setting in direct sunlight should not exceed a temperature limit well below this 775°F. If the material were allowed to anneal, it would loose its heat treated strength and this might fail under load.

Physical hazzards to the operator do not exist since he is removed from the backhoe.

FORESEEABLE DESIGN PROBLEMS

The main area of failure would have been in the area of hydraulic line heat transfer; however, this problem has already been solved by methods previously discussed. The method used to cool the fluid as it returns to the tractor has not been considered since it is not our responsibility.

The power source that drives the hydraulic pump was never specified. We simply assumed that it existed and were not concerned with the problems that would arise because of it.

The bellypack controls will be very awkward to the operator since the operator will be somewhat removed from the work area and will not have a good view. He will not be able to tell how deep the hole is nor the angle that the bucket is cutting.

We have not made any provision for the mobility of the tractor. When the trench is dug in the area that is specified, the tractor will need to move forward to dig the next portion. It will be almost impossible for the operator, which is removed from the tractor, to get it in the correct location, assuming that there are the appropriate controls added to his bellypack.

The only foreseeable solution is to install a computer guidance system that would have the trench coordinates already preprogramed in it. Then when the trench has been dug up to the base of the tractor, the computer would then advance it ten feet so that the trench could continued to be dug.

If a computer were to be incorporated into the backhoe system, it should include the actual backhoe operation. It would be ideal to be able to specify the depth of cut and cordinates of the trench and let the computer do the rest.

A backhoe like this, compared to the one we have designed is like comparing complexity to simplicity.

In my opinion, the backhoe designed with controls contained on a bellypack is not feasible, and the computerized backhoe is the only way to solve the problem of the operator being removed from the machine.

CONCLUSIONS

The lunar environment introduces numerous obstacles to be considered when designing a backhoe for the moon. The lack of air, heat dissipation, soil mechanics, and gravitational field all add to the complexity of creating a machine that will efficiently dig foundation trenches for a lunar base.

Heat dissipation appears to cause the greatest difficulties. Not only does the heat fail to convect in the lunar environment, but the electronic equipment is not designed to withstand temperatures encountered on the moon. The limited time for the electrical component's usage could be overcome by further detailed design in the electronic components used.

Continuous operation of a hydraulic system in such an environment is indeed possible using commercially available components as long as a heat sink is available. The most important considerations are preventing heat entry by radiation and providing an adequate amount of surface area for heat dissipation to the cooling fluid. These areas are critical since the backhoe is a dynamic implement and must be designed such that there is a minimum of bulk and weight. We are sure that further, more detailed analysis of the problems will yield designs that are much more efficient and effective, yet we feel that our design is a credible first approximation.

The control system that was designed is simple in form. The components used were chosen for compatibility and for simplicity of the overall system. While closed loop feedback using load sensing transducers was not used, the open loop system we chose is a workable one, with many measures to insure that the system would not be overloaded by the operator. The backhoe can be safely controlled by an operator in a space suit, but not easily. The sending unit chosen has rather small turn controls making it difficult to operate two controls at once with one hand. Certain measures were taken to overcome this problem, and the unit has been used to control a three degree-of-freedom forklift, but the operator will probably find operating the backhoe a little tedious.

The mechanical layout of the backhoe is a design that will achieve the project objectives. The C-section of the boom was neccesary to provide space for the radiators in the hydraulic circuit. Such a section will require that the boom be internally braced to prevent floppyness.

Recommendations

It is recommended that an alternative to a backhoe be considered for the earth-moving requirements, such as using a "Ditch Witch". The "Ditch Witch" potentially has several advantages over the backhoe: its rate of soil removal is greater than that of the backhoe, it can collect loose dust as it removes the soil to prevent a visibility problem for the operator, and it can be programmed to dig a ditch without an operator being present. Other specific recommendations for each section of the backhoe follow below.

Hydraulic System

Since each side of the piston displaces a different amount of fluid , the flow rate of fluid to one side of the cylinder and out of the other is not the same. To balance the flow rate between sides of the cylinders, it is recommended that counter-balanced flow rate control valves be used in the hydraulic circuit. Also, it is suggested that the design of a pneumatic backhoe be considered. Since air has no flash point, a pneumatic system would ease the maximum temperature constraint of the working fluid in the circuit.

Controls

It is recommended that the controller be redisigned to include two joysticks to allow for easier two-hand operation. Load sensing feedback control could be incorporated into the control system to provide for better system protection. Also, a computer could be used to control the backhoe in certain standard operations, such as in digging a ditch.

Mechanical Design

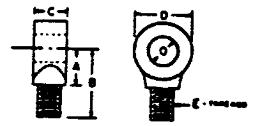
It is recommended that an existing design be used for the dimensions of the boom, bucket, and dipstick. With such a design, the kinematic calculations have been carried out, and the design has already been optimized. With backhoes of many different specifications curently available, one should be able to choose a design that would meet specific needs. Given such a design, more effort could be concentrated into the areas of heat transfer and material selection.

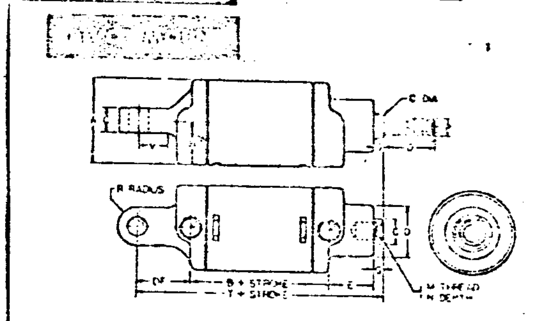
III BOOM CYLINDER

Extended Length	58"
Retracted Length	34"
Max Force $\longleftrightarrow$	8000 lb
Min Force $\rightarrow\leftarrow$	5000 lb
System Pressure	2500 lb/in²

ROD EYE

CARTER PART NO	A	B	C	D	E	O
RE 1¾ RE 2	¾	1⅝	¾	1 7/16	½-20	¾
RE 3	⅞	2	1	1 15/16	⅝-16	⅞
RE 4	[illegible]	2 7/16	1¼	2	⅞-14	1
RE 5	1¼	2⅝	1½	2½	1-14	1⅜
RE 6	1⅝	3½	1⅝	3	1¼-12	1½
RE 8	2	4⅛	2¼	3¾	1½-12	1⅝





Model DNX (non-cushioned) Model DFX (cushioned front end only)
Model DBX (cushioned both ends) Model DRX (cushioned rear end only)

1 Rod Diameter d, $N_o = 3$ $\cancel{B} \approx 2''$

$P_{CR} = 3F_{max} = 24000\text{ lb}$
C_o (end constant) $= 1$
$\ell = \text{stroke} + \cancel{B} = (24'' + 2'')$
$E_o = 3\times10^7 \frac{lb}{in^2}$
$S_y = 100{,}000\ \text{lb}/in^2$

$$d = \sqrt[4]{\frac{64 P_{CR}\ \ell^2}{\pi^3 C_o E_o}}$$

$$d = \left(\frac{64(24000\text{ lb})(26.0\text{ in})^2}{(\pi^3)(1)(3\times10^7 \frac{lb}{in^2})}\right)^{1/4} = 1.02\text{in}$$

use $\boxed{d = 1.00''}$ and RE 3 Rod End

2 Cylinder Bore D_o

Assume Friction Pressure Loss $= 10\%(P_{FL})$
Tare $= 10\%$ (from Vendor, w/ correction for low lubricity Fluid)

P_o = Cylinder Pressure $= P_{sys} - P_{FL}$
$P_o = 2250 \frac{lb}{in^2} = (2500 \frac{lb}{in^2})(1-.10)$

$$F_{total} = F_{max} + F_{tare} = 8000\text{lb}(1+.10) = 8800\text{lb}$$

$$F_{total} = P_o A_o = P_o \frac{\pi D_o^2}{4} \Rightarrow D_o = \sqrt{\frac{4F_{total}}{\pi P_o}}$$

$$D_o = \sqrt{\frac{4(8800\text{lb})}{\pi(2250 \frac{lb}{in^2})}} = 2.23\text{in}$$

use $\boxed{D_o = 3''}$

3 Find Regulator Valve Pressure P_{RV}

$$P_{RV} = \frac{F_{total}}{\pi \frac{D_o^2}{4}} = \frac{8800\text{lb}}{\pi\frac{(3.0\text{ in})^2}{4}} = 1245 \frac{lb}{in^2}$$

$$\boxed{P_{RV} = 1250 \frac{lb}{in^2}}$$

4 Stoptube Length S_o

same as II

$$\boxed{S_o = 2\tfrac{1}{8}''}$$

5 Deflection δ

where L_o = same as II

$L_o = 31\frac{7}{32}''$ $\quad \delta = \frac{F_{max} L_o}{E_o A_o}$

$$\delta = \frac{(8000\text{lb})(31.219'')}{(3\times10^7 \frac{lb}{in^2})(.7854\text{in}^2)}$$

$$\boxed{\delta = .0106''}$$

6 Summary

Extended Length	59"
Retracted Length	36"
Bore Size	3"
Piston Rod Diam	1"
Stoptubes (2) each	2 5/8"
Max. Piston Speed*	9.4 in/sec
Flow Rate (for above)*	20.3 gpm
Regulator Valve Setting	1250 lb/in²

* See Attached Chart

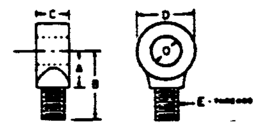
HYDRAULIC CYLINDER SPECIFICATIONS

I. BUCKET CYLINDER

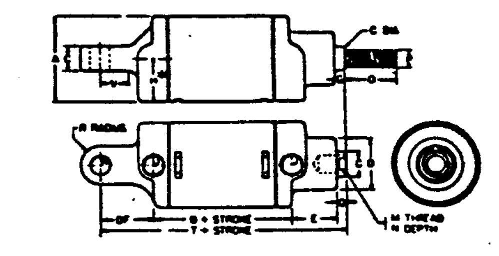
Extended Length	52"
Retracted Length	34"
Max Force $\longleftrightarrow$	3000 lb
Min. Force $\rightarrow\!\!\leftarrow$	2000 lb
System Pressure (P_{sys})	2500 lb/in^2

ROD EYE

CARTER PART NO	A	B	C	D	E	O
RE 1½ / RE 2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]-20	[illegible]
RE 3	[illegible]	2	1	[illegible]	[illegible]-16	[illegible]
RE 4	[illegible]	[illegible]	[illegible]	2	[illegible]-14	1
RE 5	[illegible]	[illegible]	1½	[illegible]	1-14	[illegible]
RE 6	[illegible]	3½	[illegible]	3	[illegible]-12	[illegible]
RE 8	2	[illegible]	[illegible]	[illegible]	1½-12	[illegible]



PIVOT MOUNT



Model DNX (non-cushioned) Model DBX (cushioned both ends) Model DFX (cushioned front end only) Model DRX (cushioned rear end only)

1 Find Rod Diameter d

Euler Formula (Buckling) $N_o = 3$

$P_{CR} = 3F_{max} = 9000\ lb$

C_o (end constant) $= 1$

$\ell = $ stroke $+ B = 18'' + 2''$ (Assume $B = 2''$)

$E_o = 3\times10^7\ lb/in^2$

$S_y = 100{,}000\ lb/in^2$ (from vendor)

$$d = \sqrt[4]{\frac{64\, P_{CR}\, \ell^2}{\pi^3 C_o E_o}}$$

$$d = \left(\frac{64\,(9000\,lb)(20\,in)^2}{\pi^3(1)(3\times10^7\ lb/in^2)}\right)^{1/4} = .705''$$

use $\boxed{d = .75''}$ use Rod End RE2 w/ $B = 1\,3/8''$

2 Find Cylinder Bore D_o

Assume Friction Pressure Loss = 10% (P_{FL})

Tare = 10% (Vendor, w/correction for low lubricity fluid)

P_o = Cylinder Pressure = $P_{sys} - P_{FL}$

$P_o = 2250\ lb/in^2 = (2500\ lb/in^2)(1-.10)$

$F_{total} = F_{max} + F_{Tare} = 3000\,lb\,(1+.10) = 3300\,lb$

$$F_{total} = P_o A_o = P_o \frac{\pi D_o^2}{4} \Rightarrow D_o = \sqrt{\frac{4F_{total}}{\pi P_o}}$$

$$D_o = \sqrt{\frac{4(3300\,lb)}{\pi(2250\ lb/in^2)}} = 1.36\,in$$

use $\boxed{D_o = 1.5''}$

3 Find Regulator Valve Pressure P_{RV}

$$P_{rv} = \frac{F_{total}}{\pi\frac{(D_o)^2}{4}} = \frac{(3300\,lb)}{\pi\frac{(1.5\,in)^2}{4}} = 1867.4\ \frac{lb}{in^2}$$

$\boxed{P_{RV} = 1900\ lb/in^2}$

4 Find Stoptube Length S_o

$S_o = (34'' - T - \text{stroke} - A)/2$

$= (34'' - 5\,3/4'' - 18'' - 5/8'')/2$

$\boxed{S_o = 4\,13/16''}$

5 Find Rod Deflection δ

where $L_o = (\text{stroke} + \frac{G}{2} + E + B + S_o)$

$L_o = (18'' - 1\,19/32'' + 1\,11/32'' + 1\,3/8'' + 7\,5/16'')$

$L_o = 29\,5/8''$ $\quad \delta = \dfrac{F_{max} L_o}{E_o A_o}$

$$\delta = \frac{(3000\,lb)(29.625'')}{(3\times10^7\ lb/in^2)(.4418\,in^2)}$$

$\delta = .0067''$

6 Summary of Results

1) Extended Length	52"
2) Retracted Length	34"
3) Bore Size	1½"
4) Piston Rod Diam.	¾"
5) Stoptubes (2) length each	5 13/16"
6) Regulator Valve Setting	1900 lb/in^2
7) Max Piston Speed*	24 in/sec
8) Flow Rate (for above)*	11 gpm

* See Attached Chart

II DIPSTICK CYLINDER

Extended Length 58"
Retracted Length 34"
Max Force ⟷ 6500 lb
Min. Force ⟶⟵ 4000 lb
System Pressure 2500 lb/in²

1 Rod Diameter d, $N_o = 3$ $\phi \approx 2''$

$P_{CR} = 3\,F_{max} = 19{,}800\ \text{lb}$
C_o (end constant) $= 1$
$\ell = \text{stroke} + \phi = (24'' + 2.0'')$
$E_o = 3 \times 10^7\ \text{lb/in}^2$
$S_y = 100{,}000\ \text{lb/in}^2$

$$d = \sqrt[4]{\frac{64\,P_{CR}\,\ell^2}{\pi^3 C_o E_o}}$$

$$d = \left(\frac{64\,(19{,}800\ \text{lb})(26.0\ \text{in})^2}{(\pi^3)(1)(3\times10^7\ \frac{\text{lb}}{\text{in}^2})}\right)^{1/4} = .97''$$

use RE3 Rod End, use $\boxed{d = 1.00''}$

2 Cylinder Bore D_o

P_{FL} (Friction Pressure Loss) = 10%
Tare = 10% (low lubricity fluid)

P_o = Cylinder Pressure $= P_{sys} - P_{FL}$
$P_o = 2250\ \text{lb/in}^2 = (2500\ \text{lb/in}^2)(1 - .10)$

$$F_{total} = F_{max} + F_{tare} = 6600\ \text{lb}(1 + .10) = 7260\ \text{lb}$$

$$F_{total} = P_o A_o = P_o \frac{\pi D_o^2}{4} \Rightarrow D_o = \sqrt{\frac{4 F_{total}}{\pi P_o}}$$

$$D_o = \sqrt{\frac{4(7260\ \text{lb})}{\pi(2250\ \frac{\text{lb}}{\text{in}^2})}} = 2.027''$$

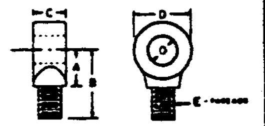
use $\boxed{D_o = 2''}$ and $\boxed{P_{RV} = 2400\ \frac{\text{lb}}{\text{in}^2}}$

3 Stoptube Length S_o

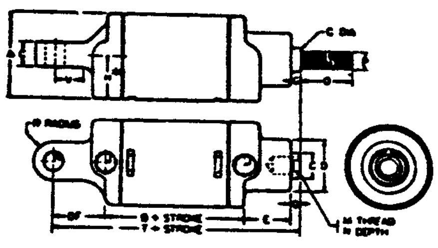
$$S_o = (36'' - T - \text{stroke} - A)/2$$
$$= (36'' - 6\tfrac{7}{8}'' - 24'' - \tfrac{7}{8}'')/2$$
$$\boxed{S_o = 2\tfrac{1}{8}''}$$

ROD EYE

CARTER PART NO	A	B	C	D	E	O
RE 1½ / RE 2	[illegible]	1 [illegible]	[illegible]	1 [illegible]	½-20	½
RE 3	[illegible]	2	1	1 [illegible]	¾-16	[illegible]
RE 4	[illegible]	2 [illegible]	1 [illegible]	2	⅞-14	1
RE 5	1 [illegible]	2 [illegible]	1½	2 [illegible]	1-14	1 [illegible]
RE 6	1 [illegible]	3½	1 [illegible]	3	1¼-12	1½
RE 8	2	4 [illegible]	2 [illegible]	3 [illegible]	1½-12	1 [illegible]



PIVOT MOUNT



Model DNX (non-cushioned) Model DFX (cushioned front end only)
Model DBX (cushioned both ends) Model DRX (cushioned rear end only)

4 Deflection δ (Piston Rod)

Where $L_o = (\text{stroke} + \frac{B}{2} + E + \phi + S_o)$

$$L_o = (24'' + 1\tfrac{27}{32}'' + 1\tfrac{11}{16}'' + 1\tfrac{3}{8}'' + 2\tfrac{5}{4}'')$$

$L_o = 31\tfrac{7}{32}''$ $\quad \delta = \dfrac{F_{max} L_o}{E_o A_o}$

$$\delta = \frac{(6500\ \text{lb})(31.219'')}{(3\times10^7\ \frac{\text{lb}}{\text{in}^2})(.7854\ \text{in}^2)}$$

$$\delta = .00861''$$

5 Summary

Extended Length	59"
Retracted Length	36"
Bore Size	2"
Piston Rod Diam.	1"
Stoptubes (2) length each	2 ⅛"
Max. Piston Speed*	13.5 in/sec
Flow Rate (for above)*	11.0 gpm
Regulator Valve Setting	2400 lb/in²

* See Attached Chart

IV SWING CYLINDERS (2)

Extended Length 20"
Retracted Length 14"
Max Force (combined) 18000 lb
System Pressure 2500 $\frac{lb}{in^2}$

1 Piston Diameter D_o

Assume Friction Pressure Loss $(P_{FL}) = 10\%$
Tare = 10% (from Vendor, w/correction for low lubricity Fluid)

P_o = Cylinder Pressure $= P_{sys} - P_{FL}$

$$P_o = 2250 \frac{lb}{in^2} = (2500 \tfrac{lb}{in^2})(1 - .10)$$

$$F_{Total} = F_{max} + F_{tare} = 18000\,lb(1 + .10) = 19{,}800\,lb$$

$$F_{Total} = P_o\, A_{Total} = P_o \left(2\frac{\pi D_o^2}{4} - \frac{\pi d^2}{4}\right)$$

assume d (Piston Rod Diam.) = 1"

$$F_{Total} = P_o \left(\frac{\pi D_o^2}{2} - \frac{\pi}{4}\right)$$

$$D_o = \sqrt{\left(\frac{F_{Total}}{P_o} + \frac{\pi}{4}\right)\frac{2}{\pi}} = \sqrt{\left(\frac{19{,}800\,lb}{2250\,\frac{lb}{in^2}} + \frac{\pi}{4}\right)\frac{2}{\pi}}$$

$D_o = 2.47"$ use $\boxed{D_o = 3"}$

2 Rod Diameter d, $N_o = 3$, $\beta \simeq 2"$

check for d = 1"

$$F_{max} = 18000\,lb \left(\frac{\frac{\pi D_o^2}{4}}{\frac{2\pi D_o^2}{4} - \frac{\pi d^2}{4}}\right)$$

$$F_{max} = 18000\,lb \left(\frac{\frac{\pi (3in)^2}{4}}{\frac{2\pi(3in)^2}{4} - \frac{\pi(1in)^2}{4}}\right)$$

$F_{max} = 9530\,lb$ $\quad P_{CR} = 3F_{max} = 28{,}590\,lb$

C_o (end const.) =
ℓ = stroke + β = (6" + 2")
$E_o = 3 \times 10^7 \frac{lb}{in^2}$
$S_y = 100{,}000\ lb/in^2$

$$d = \sqrt[4]{\frac{64\, P_{CR}\, \ell^2}{\pi^3 C_o E_o}}$$

$$d = \left(\frac{64(28{,}590\,lb)(8in)^2}{(\pi^3)(1)\,3\times 10^7 \frac{lb}{in^2}}\right)^{1/4} = .5956"$$

use $\boxed{d = 1"}$

Use RE3
Rod End

3 Regulator Valve Pressure P_{RV}

$$P_{RV} = \frac{F_{Total}}{A_{Total}} = \frac{F_{Total}}{\left(\frac{\pi D_o^2}{2} - \frac{\pi d^2}{4}\right)}$$

$$P_{RV} = \frac{19{,}800\,lb}{\left(\frac{\pi(3in)^2}{2} - \frac{\pi(1in)^2}{4}\right)} = 1483 \frac{lb}{in^2}$$

use $\boxed{P_{RV} = 1500 \frac{lb}{in^2}}$

4 Stoptube Length S_o

None Required

5 Deflection δ

where $L_o = (\text{stroke} + \frac{\beta}{2} + E + \beta + S_o)$

$$L_o = \left(6" + 1\tfrac{27}{32}" + 1\tfrac{22}{32}" + 2"\right) = 11\tfrac{17}{32}"$$

$$\delta = \frac{F_{max}\, L_o}{E_o\, A} = \frac{(9530\,lb)(11.531\,in)}{(3\times 10^7 \frac{lb}{in^2})(\frac{\pi}{4} in^2)}$$

$\delta = .00021[illegible] in$

6 Summary

Extended Length	20"
Retracted Length	14"
Bore Size	3"
Piston Rod Diam.	1"
Stoptubes	None
Max Piston Speed *	9.4 $\frac{in}{sec}$
Flow Rate (for above) *	2.03 gpm
Regulator Valve Setting	2400 $\frac{lb}{in^2}$

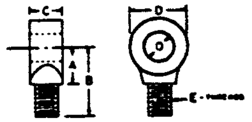
* See Attached Chart

V STABILIZER CYLINDERS (2)

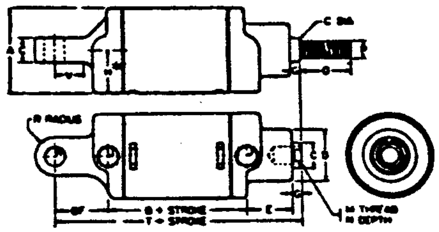
Extended Length	44"
Retracted Length	26"
Max Force $\longleftrightarrow$	13,000 lb
Min Force $\rightarrow\!\!\times\!\!\leftarrow$	Unspecified
System Pressure	2500 lb/in²

ROD EYE

CARTER PART NO	A	B	C	D	E	O
RE 1¾ / RE 2	¾	1⅝	¾	1 7/16	½-20	½
RE 3	⅞	2	1	1 11/16	¾-16	¾
RE 4	1⅛	2 7/16	1¼	2	⅞-14	1
RE 5	1⅜	2¾	1½	2¼	1-14	1⅜
RE 6	1⅝	3½	1¾	3	1¼-12	1½
RE 8	2	4¼	2¼	3¾	1½-12	1¾



PIVOT MOUNT



Model DNX (non-cushioned) Model DFX (cushioned front end only)
Model DBX (cushioned both ends) Model DRX (cushioned rear end only)

[1] Rod Diameter d, $N_o = 3$, $\not{B} \approx 2\frac{1}{2}"$

$P_{CR} = 3F_{max} = 39000\,lb$
C_o (end constant) = 1
ℓ = stroke + $\not{B}$ = (18" + 2½")
$E_o = 3\times10^7\,lb/in^2$
$S_y = 100{,}000\,lb/in^2$

$$d = \sqrt[4]{\frac{64\,P_{CR}\,\ell^2}{\pi^3 C_o E_o}}$$

$$d = \left(\frac{64(39000\,lb)(20.5\,in)^2}{(\pi^3)(1)(3\times10^7\,\frac{lb}{in^2})}\right)^{1/4} = 1.03"$$

use $\boxed{d = 1.00"}$ and RE3 Rod End with $\not{B} = 2"$

[2] Cylinder Bore D_o

Assume Friction Pressure Loss (P_{FL}) = 10%
Tare = 10% (From Vendor, w/ correction for low lubricity fluid)

P_o = Cylinder Pressure = $P_{sys} - P_{FL}$
$P_o = 2250\,\frac{lb}{in^2} = 2500\,\frac{lb}{in^2}(1-.10)$

$F_{total} = F_{max} + F_{tare} = 13000\,lb\,(1+.10) = 14{,}300\,lb$

$$F_{total} = P_o A_o = P_o \frac{\pi D_o^2}{4} \Rightarrow D_o = \sqrt{\frac{4F_{Total}}{\pi P_o}}$$

$$D_o = \sqrt{\frac{4(14{,}300\,lb)}{\pi(2250\,lb/in^2)}} = 2.844\,in$$

use $\boxed{D_o = 3"}$

[3] Find Regulator Valve Pressure P_{RV}

$$P_{RV} = \frac{F_{Total}}{\pi\frac{D_o^2}{4}} = \frac{14{,}300\,lb}{\pi\frac{(3.0\,in)^2}{4}} = 2023\,\frac{lb}{in^2}$$

$\boxed{P_{RV} = 2050\,\frac{lb}{in^2}}$

[4] Stoptube Length S_o

None Required

[5] Deflection δ

where $L_o = (\text{stroke} + \frac{B}{2} + E + \not{B} + S_o)$

$L_o = (18" + 1\frac{27}{32}" + 1\frac{22}{32}" + 2")$

$L_o = 23\frac{17}{32}"$

$$\delta = \frac{F_{max}\,L_o}{E_o\,A} = \frac{(13{,}000\,lb)(11.5?\,in)}{(3\times10^7\,\frac{lb}{in^2})(\frac{\pi}{4}\,in^2)}$$

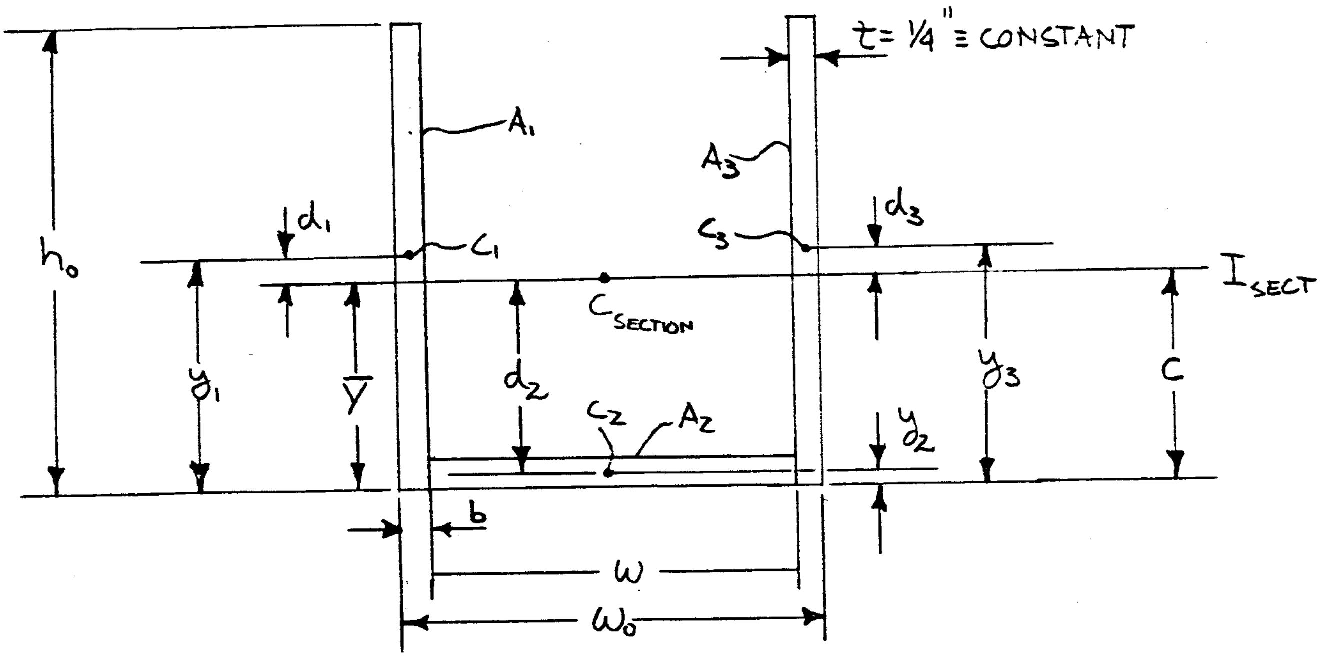
$\delta = .0064\,in$

[6] Summary

Extended Length	44"
Retracted Length	26"
Bore Size	3"
Piston Rod Diam.	1"
Stoptubes	None
Max. Piston Speed*	9.4 in/sec
Flow Rate (for above)*	20.3 [illegible]
Regulator Valve	2050 lb/in²

DETERMINING SECTION MODULUS - BOOM

USING C-SECTION - 1/4" THICKNESS
ALUMINUM : ASSUME σ_{ALLOW} = 10,500 psi



DETERMINE $C_{SECTION}$

A_i	AREA (in^2)	y_i (in)	$A_i y_i$ (in^3)
A_1			
A_2			
A_3			

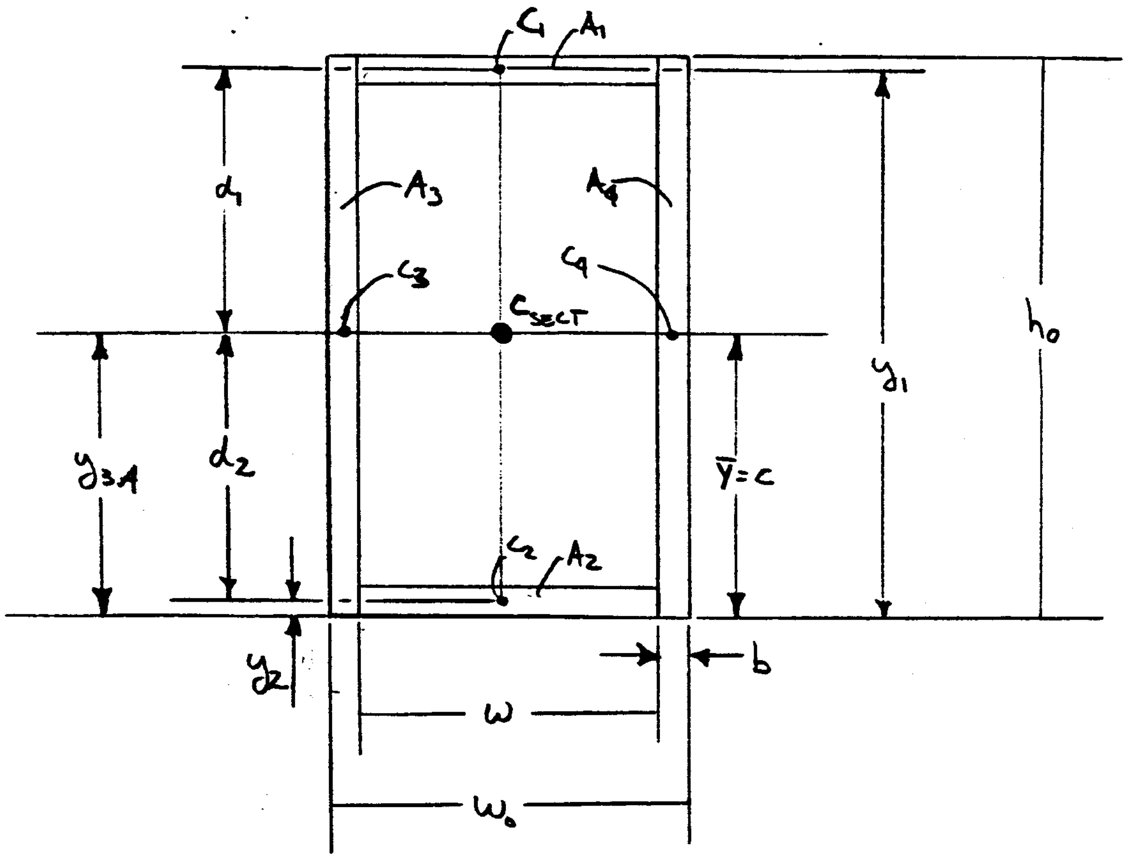
$\sum A_i$ $\qquad\qquad$ $\sum A_i y_i$

$$\Longrightarrow \quad \bar{Y} = \frac{\sum A_i y_i}{\sum A_i}$$

$\bar{Y} = c$

DETERMINING SECTION MODULUS - DIPPERSTICK

USING TUBE SECTION - 1/4" THICKNESS
ALUMINUM : ASSUME σ_{ALLOW} = 10,500 psi



$C_{SECTION}$ OF TUBE IS IN THE CENTER DUE TO SYMETRY

$A_1 = A_2$
$A_3 = A_4$

DETERMINE I_{SECT}

ALL AREAS ARE DIVIDED INTO RECTANGULAR AREA

$$I_{A_i} = \frac{1}{12} b_i h_i^3 + A_i d_i^2$$

$$I_{SECT} = \sum I_{A_i}$$

SECTION MODULUS

$$S = \frac{I_{SECT}}{c}$$

CRITERION: $S = \frac{I_{SECT}}{c} \geq \frac{M_{MAX}}{\sigma_{ALLOW}}$ WHERE

M_{MAX} ≡ MAXIMUM BENDING MOMENT FOR SECTION FROM MOMENT DIAGRAM

σ_{ALLOW} ≡ MAXIMUM ALLOWABLE STRESS GIVEN : 10,500 psi

SECTION MODULUS - BOOM

@ BOOM & DIPPERSTICK CYLINDER

COMMON PIN CONNECTION (LOCATION A)

CHOOSE $h_o = 15''$, $w_o = 6''$

A_i	AREA (in^2)	y_i (in)	$A_i y_i$ (in^3)
A_1	3.75	7.5	28.125
A_2	1.25	.125	.15625
A_3	3.75	7.5	28.125

$\sum A_i = 8.75 \text{ in}^2$ $\qquad \sum A_i y_i = 56.40625 \text{ in}^3$

$\bar{Y} = c = 6.4464$

$I_{SECT} = I_{A_1} + I_{A_2} + I_{A_3}$

$I_{A_1} = I_{A_3} = \frac{1}{12} bh^3 + A_i d_i^2$

$I_{A_1} = I_{A_3} = 74.4753 \text{ in}^4$

$I_{A_2} = 48.00 \text{ in}^4$

$I/c = 31.785 \text{ in}^3$

BM @ A = 5500 ft-lbf = 66000 in-lbf

$\sigma_{ALLOW} = 10{,}500$ psi

$M/\sigma_{ALLOW} = 6.2857 \text{ in}^3$

THUS $I/c > M/\sigma_{ALLOW}$

CHOOSE $h_o = 15''$

SECTION MODULUS - DIPPERSTICK

@ LOCATION A

CHOOSE $h_o = 4$ in $\quad W_o = 4$ in $\quad M = 1000$ ft-lbf $= 12{,}000$ in-lbf

$M_{MAX}/\sigma_{ALLOW} = 1.1429 \text{ in}^3$

$\bar{Y} = c = 2.0 \text{ in}$

$A_1 = A_2 = 1.0 \text{ in}^2$

$A_3 = A_4 = .875 \text{ in}^2$

$I_{A_1} = 1.3333 \text{ in}^4$

$I_{A_2} = I_{A_1}$

$I_{A_3} = 3.08 \text{ in}^4$

$I_{A_4} = I_{A_3}$

$I_{SECT} = 8.828 \text{ in}^4$

$I_{SECT}/c = 4.419 > M_{MAX}/\sigma_{ALLOW}$ CHOOSE $h_o = 4$ in

@ LOCATION B

$M_{MAX} = 96\,000 \text{ in-lbf}$

$M/\sigma_{ALLOW} = 9.1429 \text{ in}^3$

→ CHOOSE $h_o = 9"$ $\quad I_{SECT}/c = 14.1952 \text{ in}^3$ ✓

$\bar{Y} = c = 4.5"$

@ LOCATION C

$M_{MAX} = 73440 \text{ in-lbf}$

$M/\sigma_{ALLOW} = 6.99 \text{ in}^3$

→ CHOOSE $h_o = 6"$ $\quad \bar{Y} = c = 3"$

$I_{SECT}/c = 7.825 \text{ in}^3$ ✓

PIN DIAMETER CALCULATION

PIN MATERIAL: AISI 4340 DRAWN 1000°F.

$S_u = 182000$ PSI

$S_e' = 19.2 + .314\, S_u$

$S_e' = 76{,}348$ PSI

Surface Finish: Ground; $k_a = .89$

Reliability: .99999 $k_c = .659$

TEMPERATURE EFFECTS:

$T_{max} < 840°F$ $k_d = 1$

Stress-concentration Effects: None

$k_e = 1$

$$S_e = (.89)(.659)(76{,}348)\ \text{PSI}$$
$$= 44779\ \text{PSI}$$

$$A = \frac{F_R}{S_e * \eta * N} = \frac{\pi}{4} d^2$$

$N = 2$ .... since two shear Regions

$\eta = 6$ .... factor of safty

$$d_{min} = \frac{4 F_R}{(\pi)(S_e)(\eta)(N)} = \frac{F_R}{105{,}508} \quad \text{inches}$$

S_u = ULTIMATE TENSILE STRENGTH

S_e = Endurance limit

k_a = Surface factor

k_c = reliability factor

k_d = temperature factor

K_e = stress concentration factor

F_R = Resultant Force

η = Factor of safety

N = number of Shear Regions

d = Pin diameter

FORCE ANALYSIS

NOMENCLATURE

BDF ≡ BUCKET DIGGING FORCE - CASE A LOAD
BCF ≡ BUCKET CYLINDER FORCE
BHP' ≡ PIN REACTIONS ON BUCKET
BHP'' ≡ PIN REACTIONS ON DIPPERSTICK
DDF ≡ DIPPERSTICK DIGGING FORCE - CASE B LOAD
DCF ≡ DIPPERSTICK CYLINDER FORCE
DHP' ≡ PIN REACTIONS ON DIPPERSTICK
DHP'' ≡ PIN REACTIONS ON BOOM
DHPB ≡ PIN REACTIONS RESOLVED INTO BOOM COORDINATES
BOOM CF ≡ BOOM CYLINDER FORCE
BHP' ≡ PIN REACTIONS ON BOOM
BCM ≡ MOMENT INDUCED DUE TO BUCKET CYLINDER FORCE
BOOM CM ≡ MOMENT INDUCED DUE TO BOOM CYLINDER FORCE
DCM ≡ MOMENT INDUCED DUE TO DIPPERSTICK CYLINDER FORCE

FORCE ANALYSIS - BUCKET

CASE A LOADING: BDF = 1000 lb_f @ 50°

$+\circlearrowleft \sum M_{BHP} = 0$:

$BDF(30'') = BCF(10.125'')$

$BCF = 2963\ lb_f \qquad = \qquad 3000\ lb_f$

$\xrightarrow{+} \sum F_x = 0$:

$-BHP_x' + BCF_x + BDF_x = 0$

$BHP_x' = BCF_x + BDF_x$

$BHP_x' = 3639\ lb_f \qquad = \qquad 3640\ lb_f$

$+\uparrow \sum F_y = 0$:

$-BHP_y' - BCF_y + BDF_y = 0$

$BHP_y' = BDF_y - BCF_y$

$BHP_y' = 609\ lb_f \qquad = \qquad 610\ lb_f$

CASE B LOADING: DDF = 1000 lb_f @ 78°

$+\circlearrowleft \sum M_{BHP} = 0$:

$BCF = 2629.6\ lb_f \qquad = \qquad 2650\ lb_f$

$\xrightarrow{+} \sum F_x = 0$:

$BHP_x' = 3203.8\ lb_f \qquad = \qquad 3205\ lb_f$

$+\uparrow \sum F_y = 0$:

$BHP_y' = 821.1\ lb_f \qquad = \qquad 825\ lb_f$

CASE A : BCM = $BCF \times \cos 3° \times 13\ in \times 1\ FT/12\ in = 3250\ ft\text{-}lb_f$

CASE B : BCM = $\qquad = 2870\ ft\text{-}lb_f$

FORCE ANALYSIS - DIPPERSTICK

CASE A LOADING

$$\circlearrowleft^{+} \sum M_{DHP} = 0$$

$$-DCF(13.5'') + BCF(13.5'') + BHP_y''(63'') = 0$$

$DCF = 5847 \text{ lb}_f$ $\quad = 5850 \text{ lb}_f$

$DCM =$ $\quad = 755 \text{ ft-lb}_f$

$$\xrightarrow{+} \sum F_x = 0$$

$$-DHP_x' + DCF_x - BCF_x + BHP_x'' = 0$$

$DHP_x' = 2645 \text{ lb}_f$ $\quad = 2650 \text{ lb}_f$

$$+\uparrow \sum F_y = 0$$

$$-DHP_y' + DCF_y + BCF_y + BHP_y'' = 0$$

$DHP_y' = 6264 \text{ lb}_f$ $\quad = 6265 \text{ lb}_f$

CASE B LOADING

$$\circlearrowleft^{+} \sum M_{DHP} = 0 :$$

$DCF =$ $\quad = 6500 \text{ lb}_f$

$DCM =$ $\quad = 835 \text{ lb}_f$

$$\xrightarrow{+} \sum F_x = 0 :$$

$DHP_x' =$ $\quad = 2785 \text{ lb}_f$

$$+\uparrow \sum F_y = 0 :$$

$DHP_y' =$ $\quad = 7075 \text{ lb}_f$

CASE A

First Resolve

$DHP''_{x,y}$ into $DHPB_{x,y}$

$$DHPB_x = DHP''_y \cos 22^\circ + DHP'' \cos 68^\circ$$
$$DHPB_y = DHP''_y \sin 22^\circ - DHP'' \sin 68^\circ$$

$$DHPB_x = 6805\ lb_f$$
$$DHPB_y = -115\ lb_f$$

$\circlearrowleft + \ \Sigma M_{DHP} = 0 \qquad DCF(10.5'') - BOOM\ CF(13.5'') + DHPB_y(90'') = 0$

Boom CM = 3785 ft·lb_f

$BOOM\ CF = 3783\ lb_f = 3785\ lb_f$

$\xrightarrow{+} \Sigma F_x = 0 \qquad -BHP'_x + BOOM\ CF_x - DCF_x + DHPB_x$

$BHP'_x = 4000\ lb_f$

$+\uparrow \Sigma F_y = 0 \qquad BHP'_y - BOOM\ CF_y - DCF_y + DHPB_y$

$$\underline{BHP'_y = 245\ lb_f}$$

CASE A

BOOM CM = 3785

DCM = 5850

CASE B Resolve Into Pin Reaction on Boon

$DHPB_x = 1045\ lb_f$

$DHP_y = 70\ lb_f$

$\Sigma M_{BHP} = 0$

$BOOM\ CF = 5525\ lb_f$

$\Sigma F_y = 0$

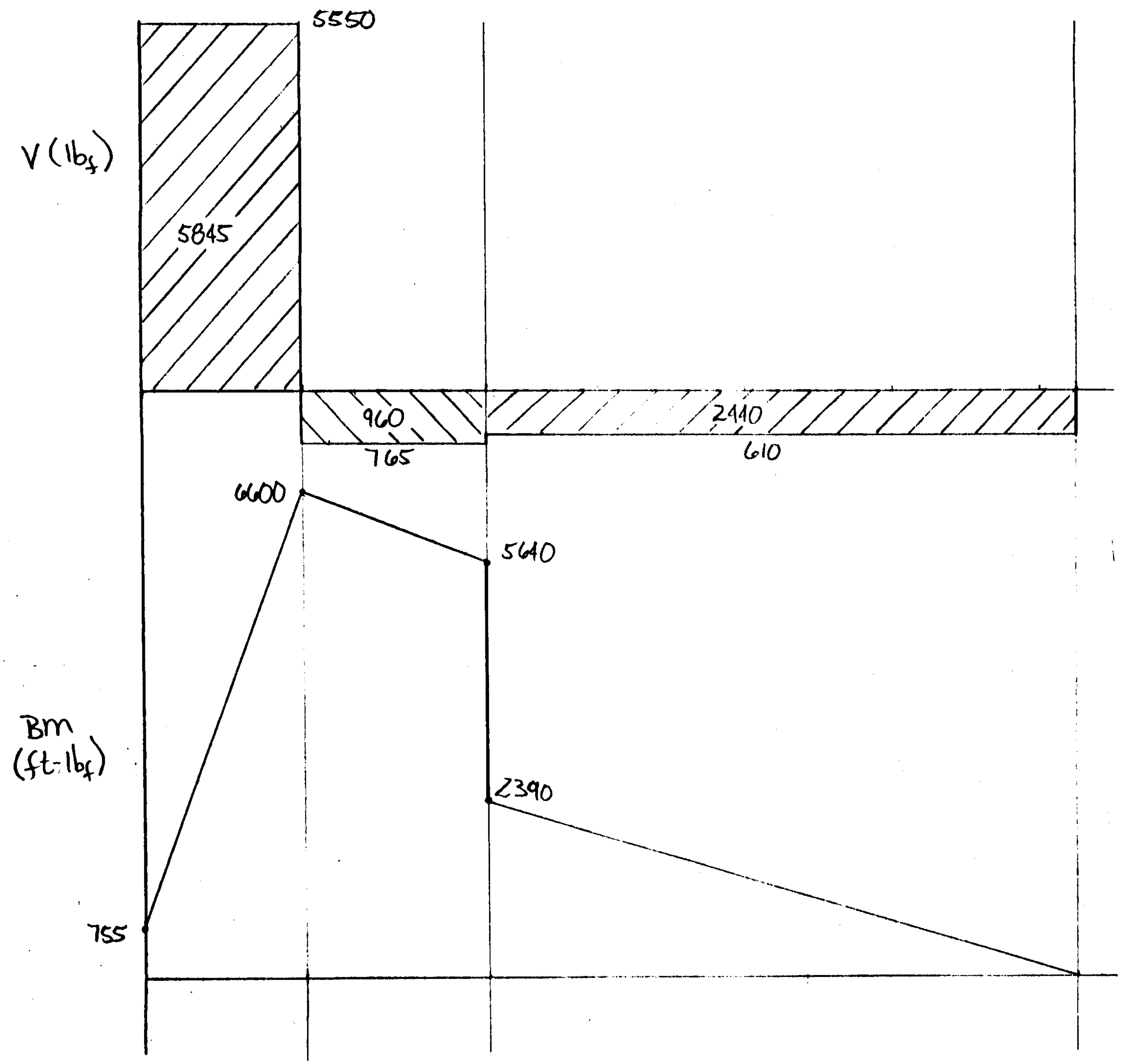
$BHP'_x = 70\ lb_f$

$\Sigma F_y = 0$

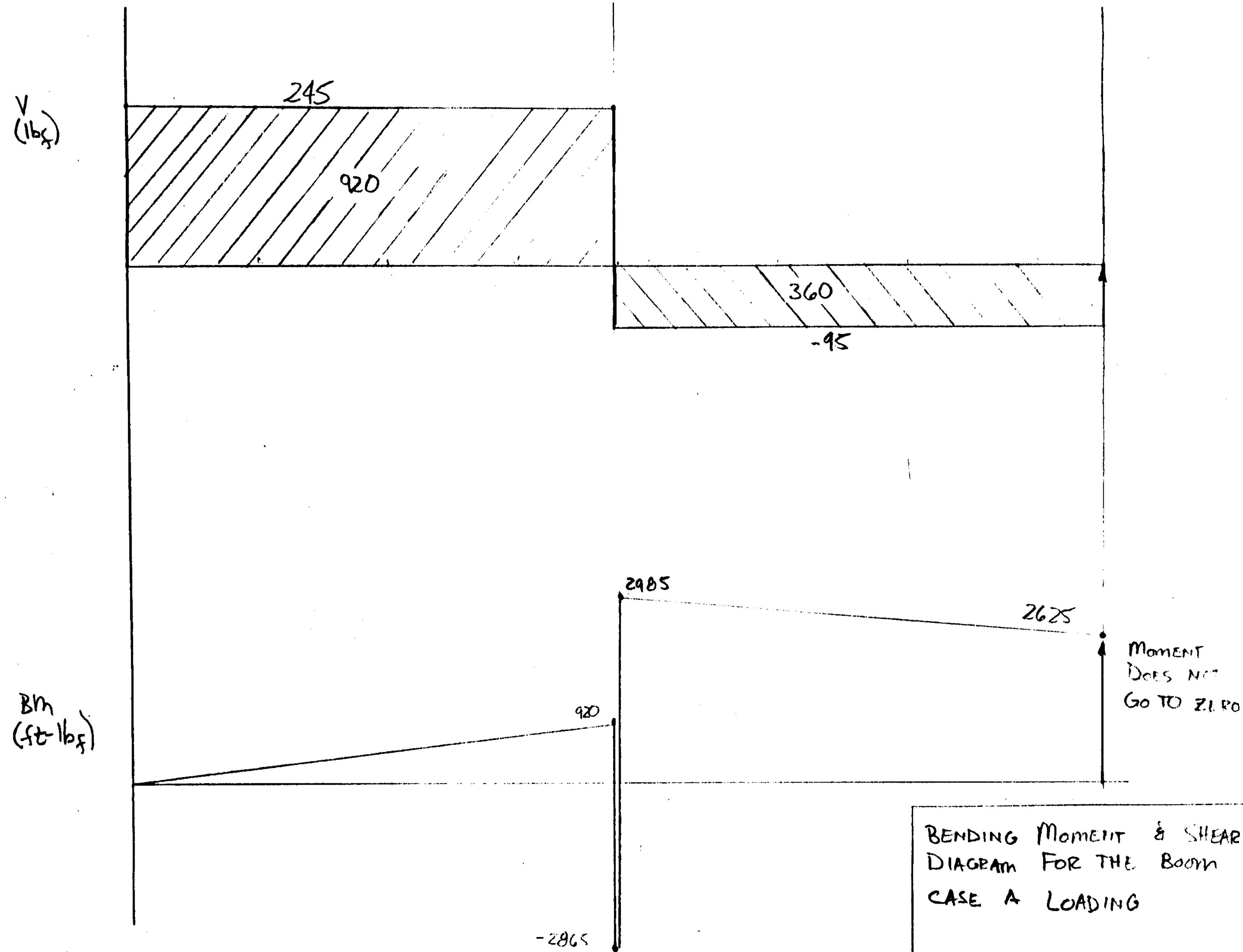
$BHP'_y = 350\ lb_f$

BOOM CM = 5525

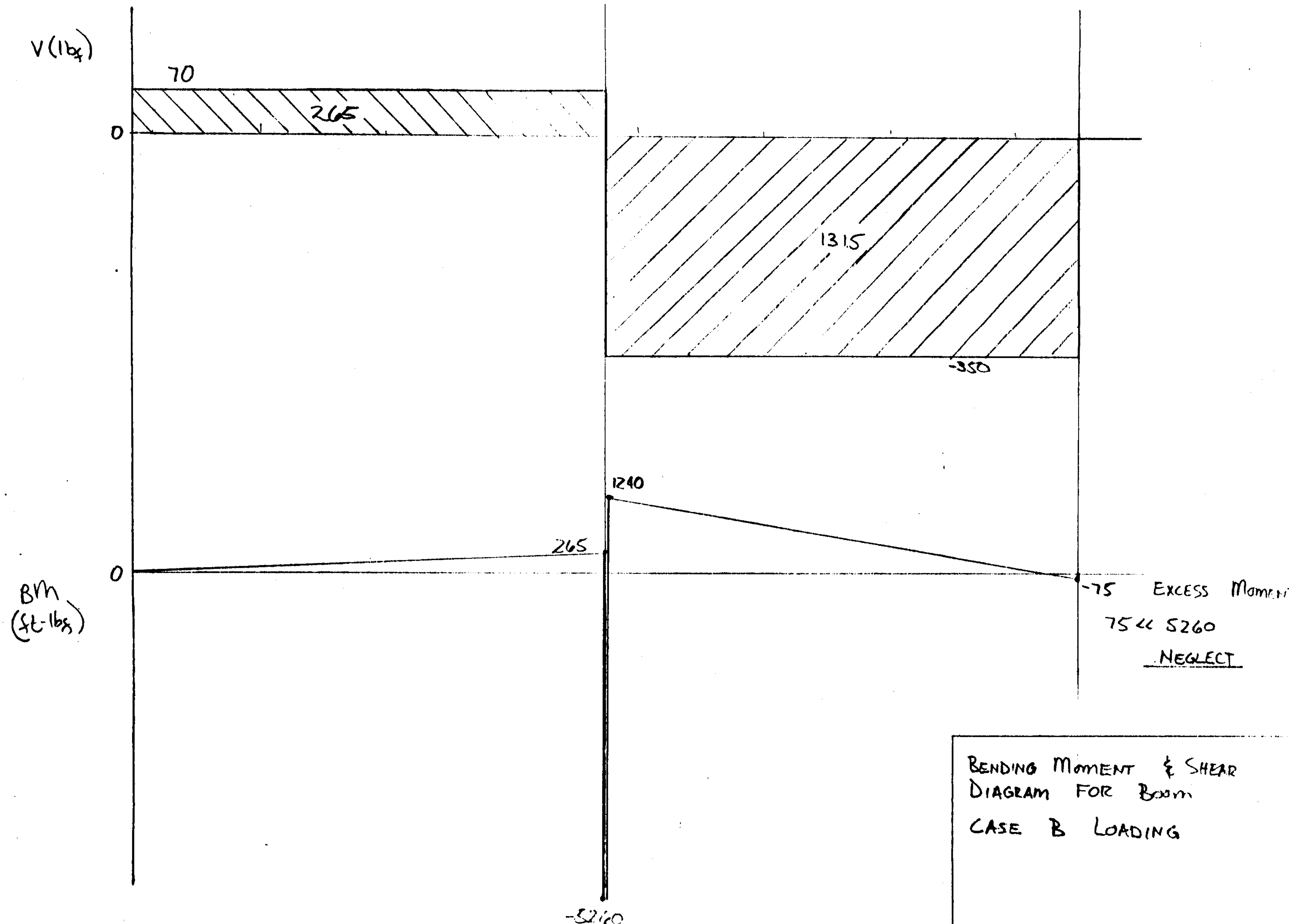
DCM = 6500



SHEAR & BENDING
MOMENT DIAGRAM
FOR DIPPERSTICK –
CASE A LOADING

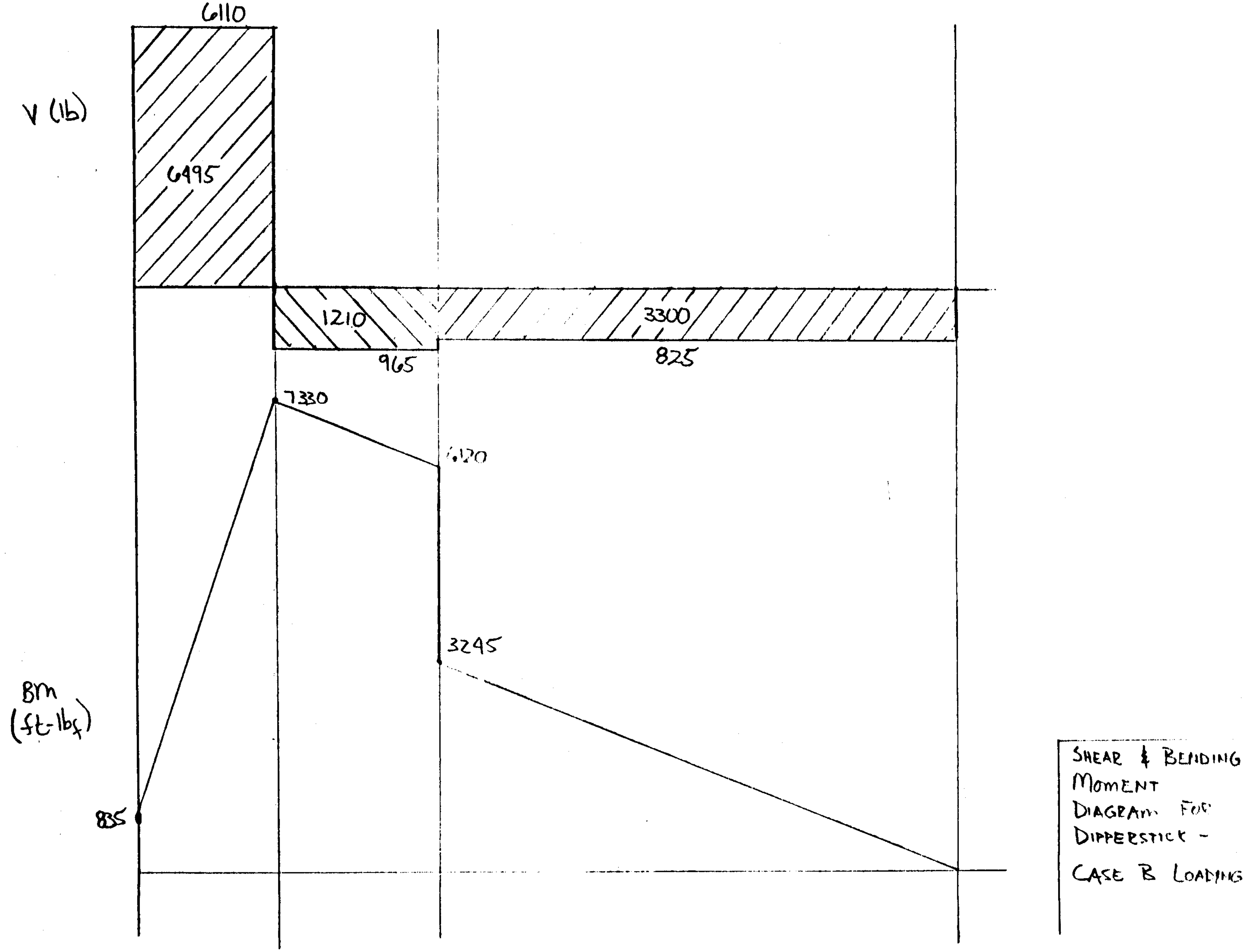


BENDING MOMENT & SHEAR DIAGRAM FOR THE BOOM
CASE A LOADING



BENDING MOMENT & SHEAR DIAGRAM FOR BOOM

CASE B LOADING


6110
V (lb)
6495
1210
3300
965
825
7330
6120
3245
BM
(ft-lbf)
835
SHEAR & BENDING
MOMENT
DIAGRAM FOR
DIPPERSTICK -
CASE B LOADING

ALTERNATE DESIGNS

Although the backhoe is functionable, it is very inefficient. The purpose of it was rather ambiguous; however, now to our understanding, it is to dig the footings of a building to be used as a lunar base. If this assumption is correct, a backhoe is the wrong machine to use. It is unnecessarily large, heavy, and clumsy. It is capable of digging holes ten feet deep which are not needed. It has a hard time digging a trench with a constant depth due to the controls. If the purpose of the machine is to dig trenches then the machine to be used is a trencher, since it would be simpler to make and also to operate.

It would be composed of two rotating wheels each with teeth about them. The two wheels would be rotating in opposite directions thus almost eliminating the horizontal digging forces. The only force of significance would be the vertical force pushing down and would be limited by the machine's weight. This force could easily be increased by accumulating the soil being dug into a large container until the desired vertical digging force is reached.

The machine would not have wheels but rather bulldozer type tracks to assure good traction.

This machine would be much easier to be computer controlled compared to the backhoe since it has such a simple design comparitively.

The first drawing shows the two rotars having their height controlled by hydraulic cylinders. For regular trenching, the depth of the first rotar (d_1) is half of that of the second (d_2).

$$2d_1 = d_2$$

Weekly Progress Reports

Design Section: Rob Schenk
Rick Wilkins

Week	Progress
1	Picked team members Picked study topic- soil mechanics Picked design topic- Backhoe
2	Group meeting Divided up study topic into sub-topics: books, technical reports, abstacts, govt. documents. Group meeting Organized material for progress meeting. Progress Meeting Suggeted a photocopy list of all material for central information file.
3	Group meeting topic is finished- have lists typed up Group meeting Gathering of all typed material for fianl listing in information file progress meeting turned in soil mechanics material started backhoe design Brazell gave us a starting point Split up into groups- controls hydraulics design Vendor informations obtained- Vermeer
4	meeting with Jeff Kreuger discussed kinematic problems and design ideas group meeting- Rob Schenck- more info on catepillar Steve-check out lunar atmosphere Scott and Brian- controls Rick- lunar soil continued with kinematic initial design Progress meeting discussed syllabus for remainder of quarter availability of IBM- Jerry Insilia Word Processing for writing report Problem statement due 5/2
5	Group meeting Discussed hydraulic line problems and design Posed problem statement for review, discussion, revision, etc. Determined capacity and digging depth final form of statement to be typed up Wednesday-Brian Rob-talk to Catapillar representative-get details on interface for hitch, hitch design initiated. Controls info received

Progress meeting
Turned in problem statement, revised digging forces.
Concentrated on hydraulic lines, shielding temperature limits, etc.
Discussed purpose of VSMF catalog
Consult this for Vendor information
controls group interface with hydraulics.
Group meeting
More vendor info- Ford
Design is complete, meet with Brazell to determine member sections.
More work on controls and valves for hydraulic system.
Consult VSMF standards to determine position for digging forces.

6 Progress meeting
early meeting with Mr. Brazell
Method for determining section modulus in backhoe sections.
Quick disconnect idea is discussed.
Control problem discussed- flow rates proportional to forces which in turn determines cylinder sizes.
Looked at buckling problems.
Materials for suitable use is suggested: low emmissivity. low reflectivity
Still need final details on hitch.
Stabilizers and swing ranges and sizes.
Choose digging forces to initiate sizing of system.

7 Materials chosen. Aluminum 6066 T-6
Modified hitch to incorporate labor intensive work.

8 Progress meeting.
Initial hitch design complete- modify for swing cylinders
Dig Forces- take individual cases to determine reaction forces, don't add
interface- choose design.
design complete, finish sections

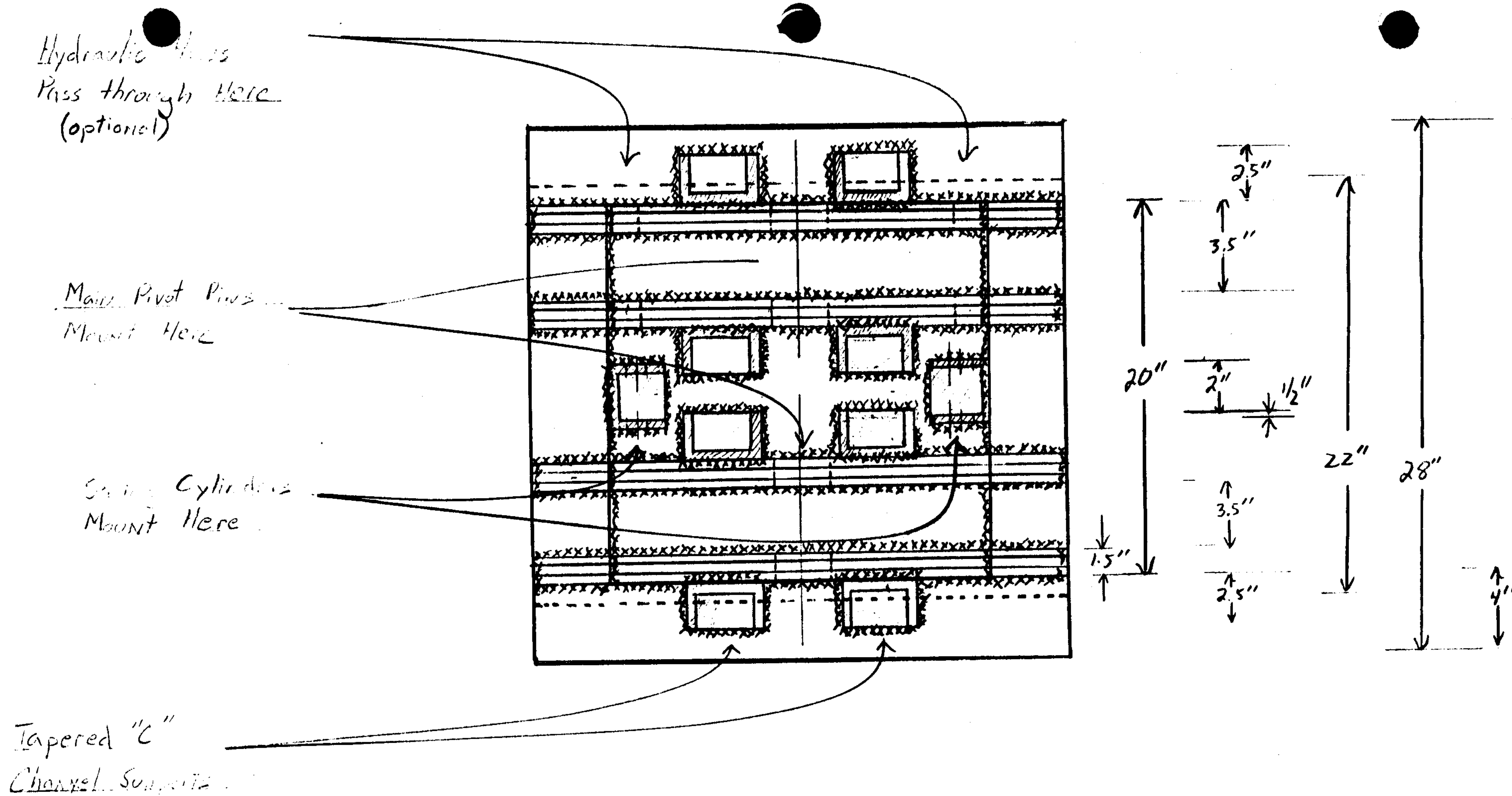
9 Initial design has to be changed- bending moments didn't work out in boom member.
Final changes made in hitch
final drawings competed.
Backhoe final drawings done.
Force analysis complete.
Choose section modulus to incorporate cylinders and heat exchanger.
Change from original tube section to C-section.

Weekly Progress Reports

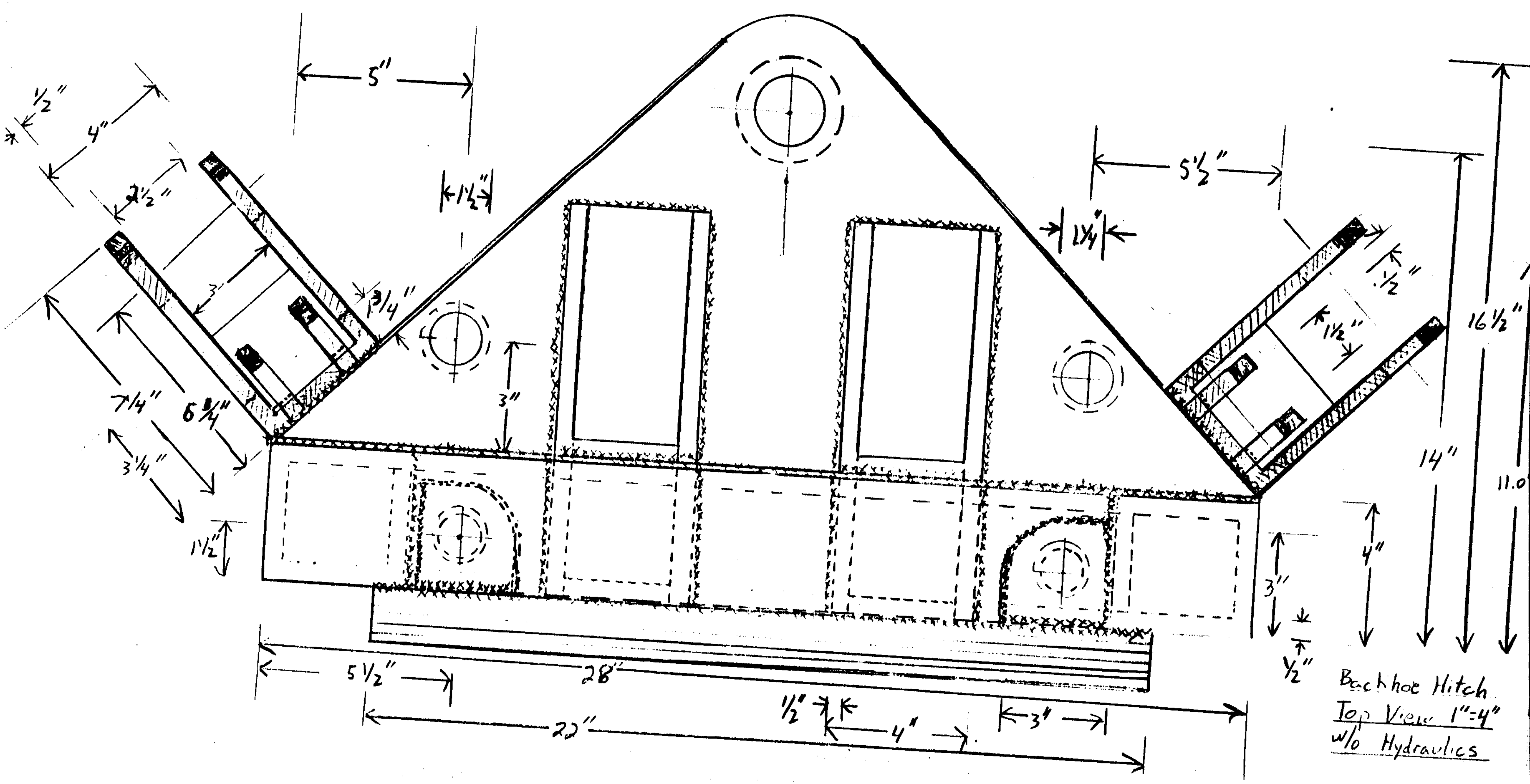
Controls Section: Brian Hatchell
Scott Landers

Week	Progress
1	Had organizational meeting to determine individual duties for research assignment: Hatchell, technical reports Landers, government reports
2	Searched microfiche, journals, and government reports relevant to soil mechanics.
3	Continued searching library material for soil mechanics literature relevant to project. Wrote down findings to be compiled into class data base.
4	Researched TomCat for vendors involved in remote control, radio control, and electro-mechanical hydraulic valves. Considered alternative designs to relieve the operator's inherent vision problem. Decided that radio control was the safest solution to the problem.
5	Worked on problem statement, and researched Tomcat for additional vendors that might be needed. Wrote letters to 15 prospective venders and called 3 others in the areas of hydraulic valves and radio control.
6	Wrote up problem statement, typing. Called up Moog for servo valves.
7	Researched library for textbooks that could help in setting up the control system. Selected 5 books that were moderately helpful. Read these books and learned about the specific kinds of valves that would be needed to complete the system. Learned about overall hydraulic system. Group meeting. Looked over the vendor information that we received and determined that we probably had information on all the components that we would need, except for a radio receiver and a transmitter. Called up the Moog representative for the Atlanta area and he said that Moog had a radio receiver and transmitter that would provide the proportional control of servo-valves that we would require. He agreed to send us the information that we would need to set the system up.
9	Given the hydraulic information from Rick and Steve, we were able to determine the valves that would be required. Luckily, we had a stackable control valve that could be easily set up on the backhoe. The sizes and accessories that would be needed were determined. Considered cross-port verus regular relief valves and decided that the regular relief valve was the best for our application. Went out to the Ford tractor place and looked at a

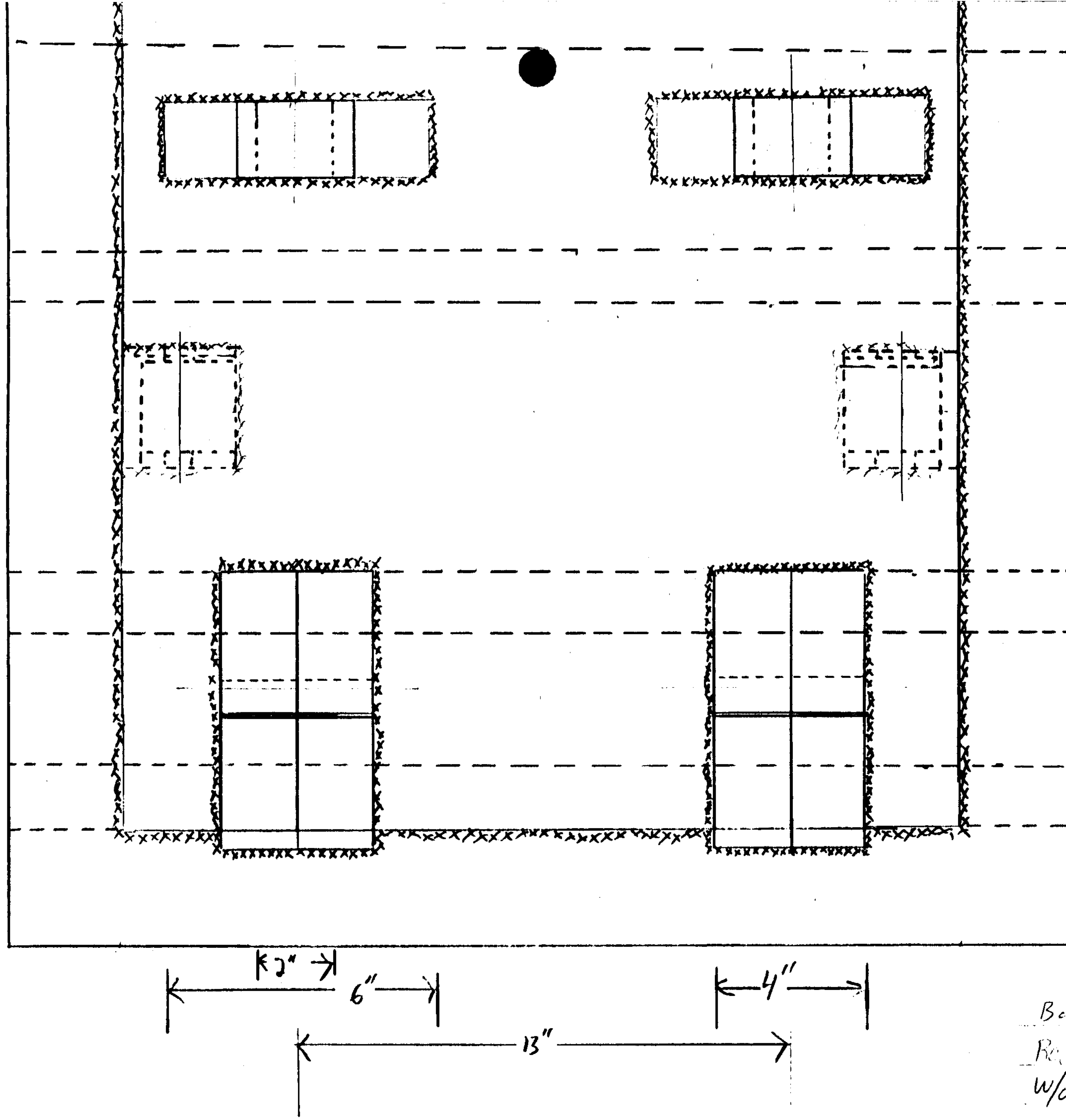
backhoe similar to the backhoe that we were designing. Because the Moog representative failed to mail us the information that we asked for, we drove out to his office. We talked with him for about 45 minutes about the system that we were designing. He suggested several alternatives including position and load feedback control. These measures would require additional electronic hardware, and we decided that using relief valves and pressure controlling valves would be a simpler, but still adequate solution.
Completed the overall design of the control system.
Worked on typing up the report.



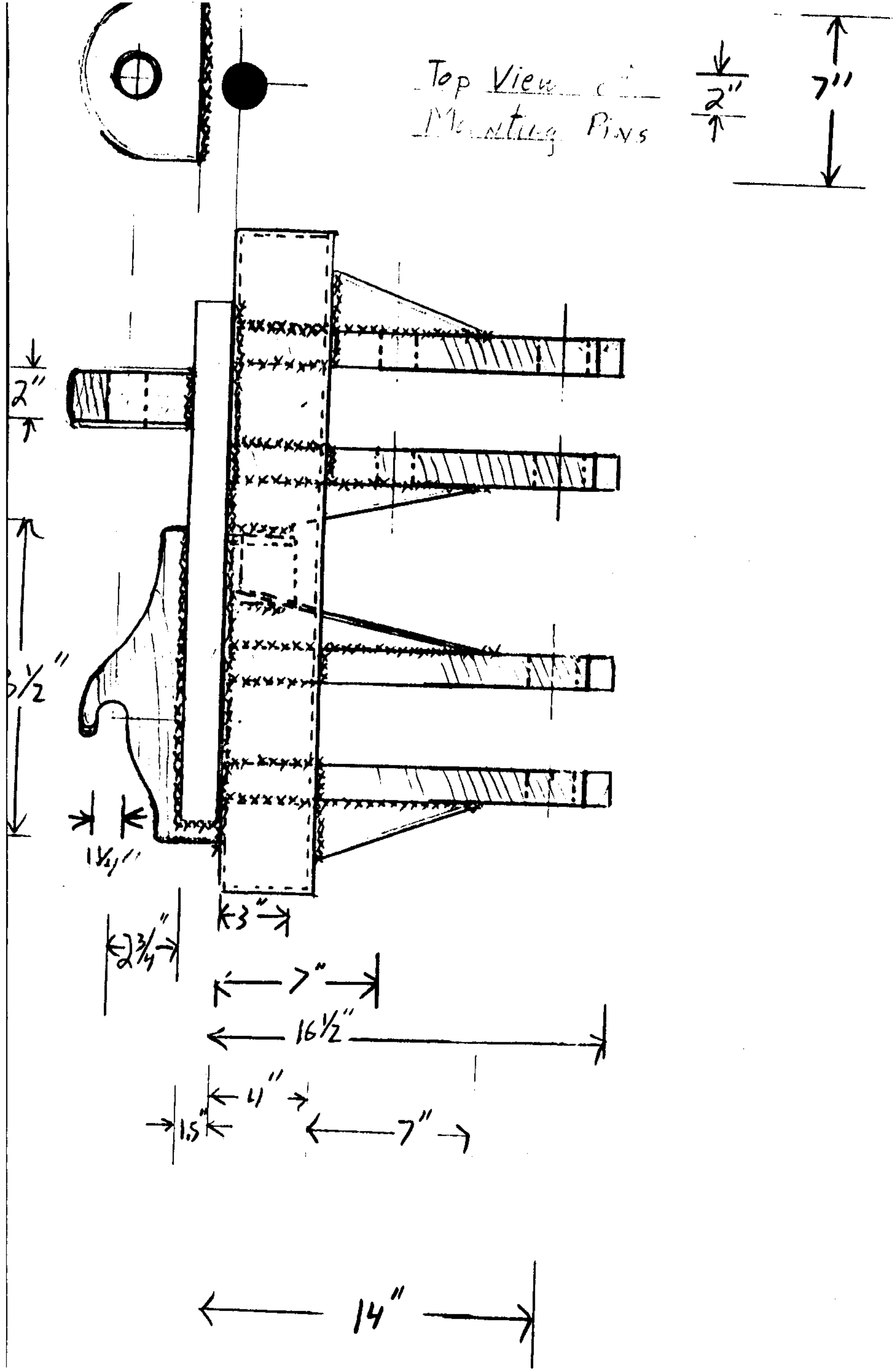
Backhoe Hitch
Front View 1" = 8"



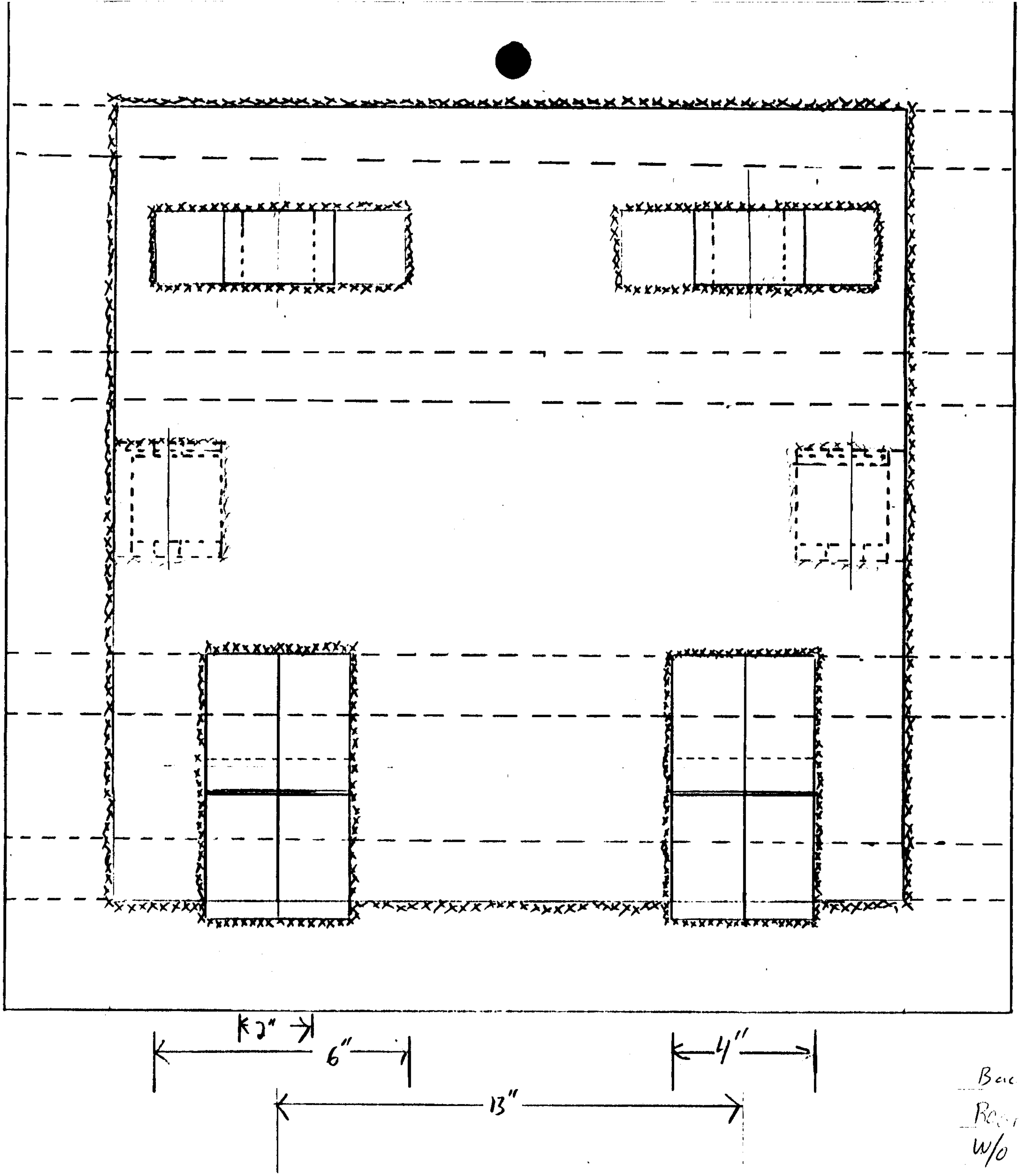
Backhoe Hitch
Top View 1"=4"
w/o Hydraulics



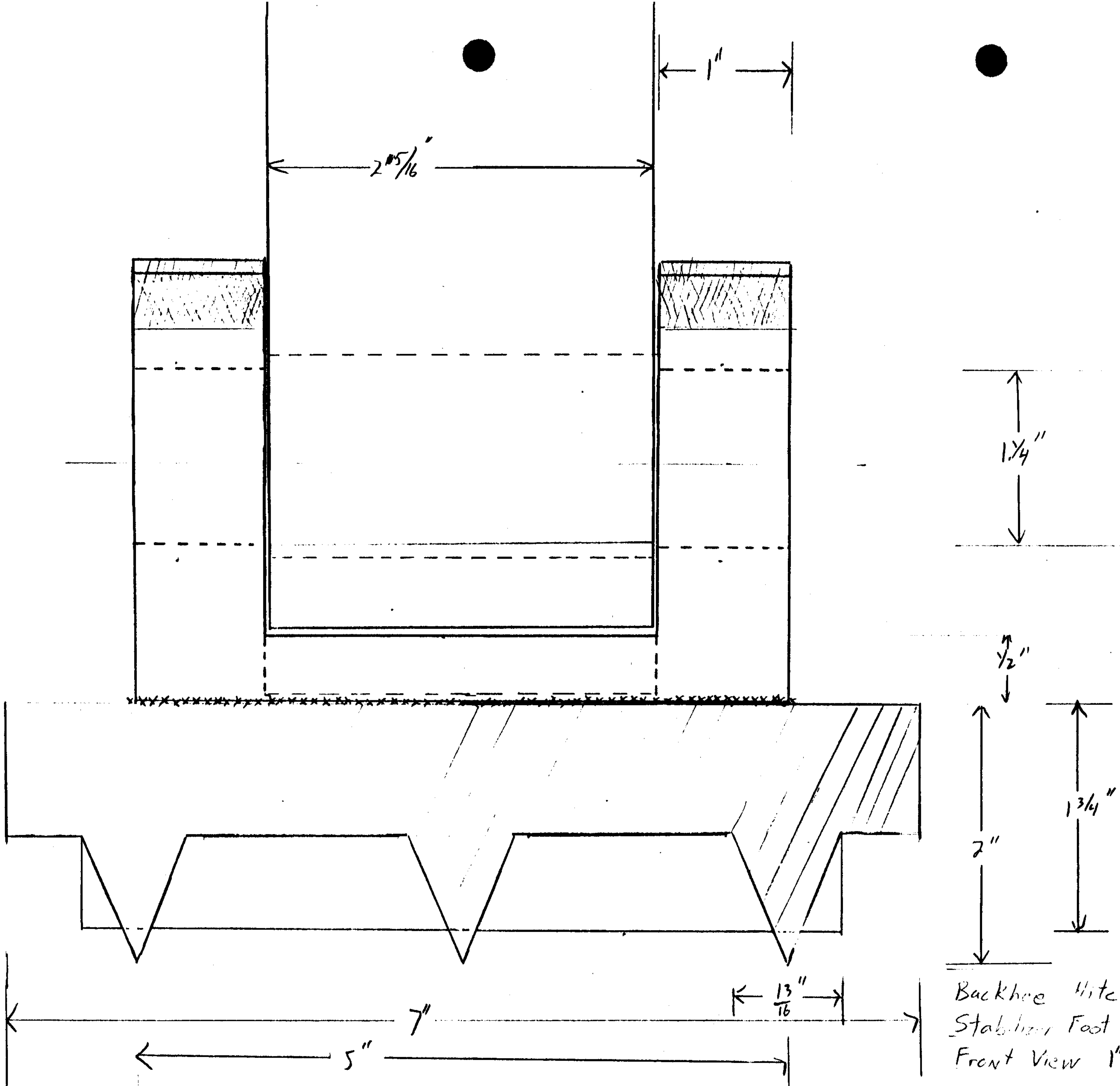
Backhoe Hitch
Rear View 1" : 4"
w/o Cylinder Supports
hydraulics, stabilizers



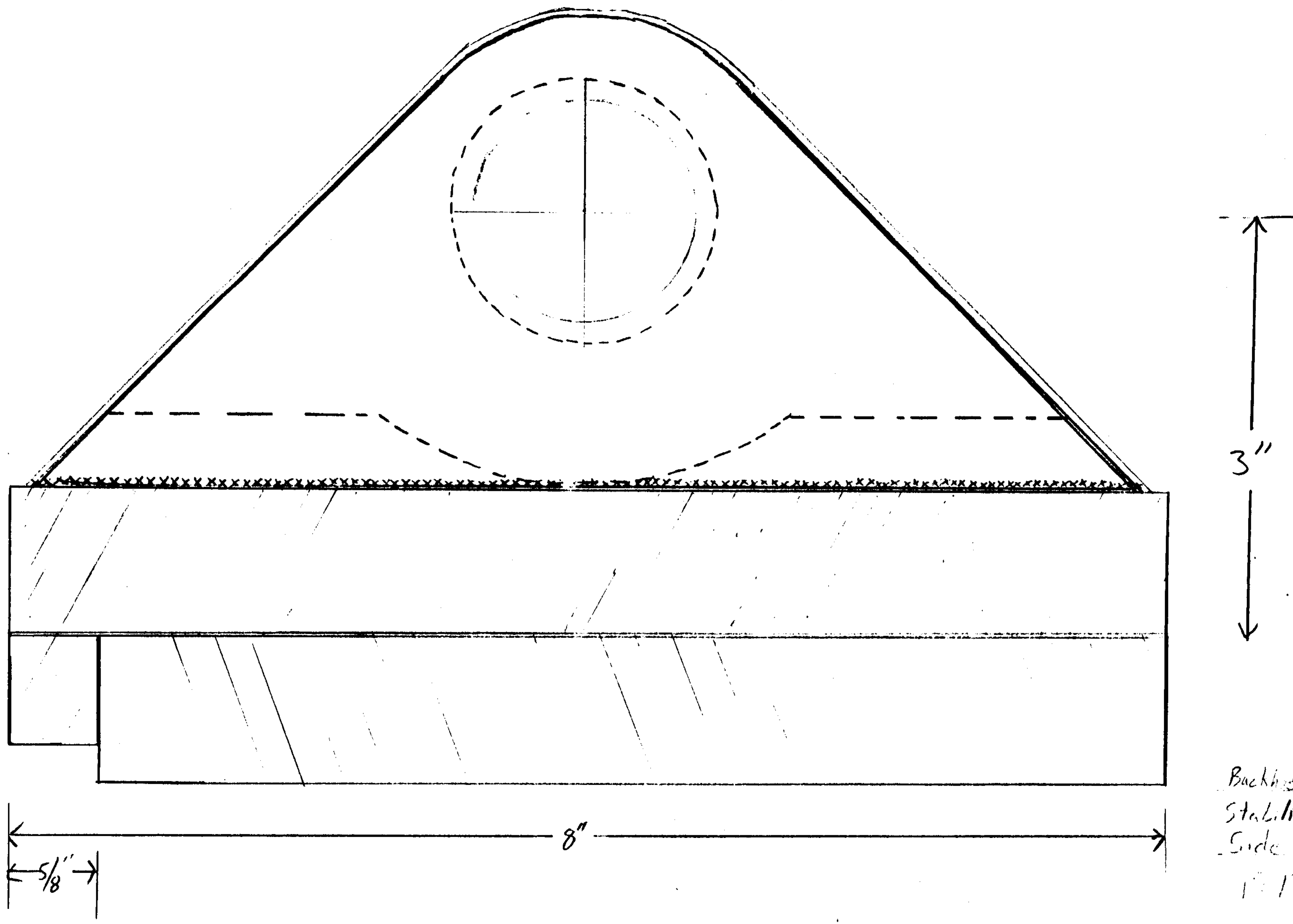
Backhoe Hitch
Side View 1" = 8"
w/ Hydraulics or



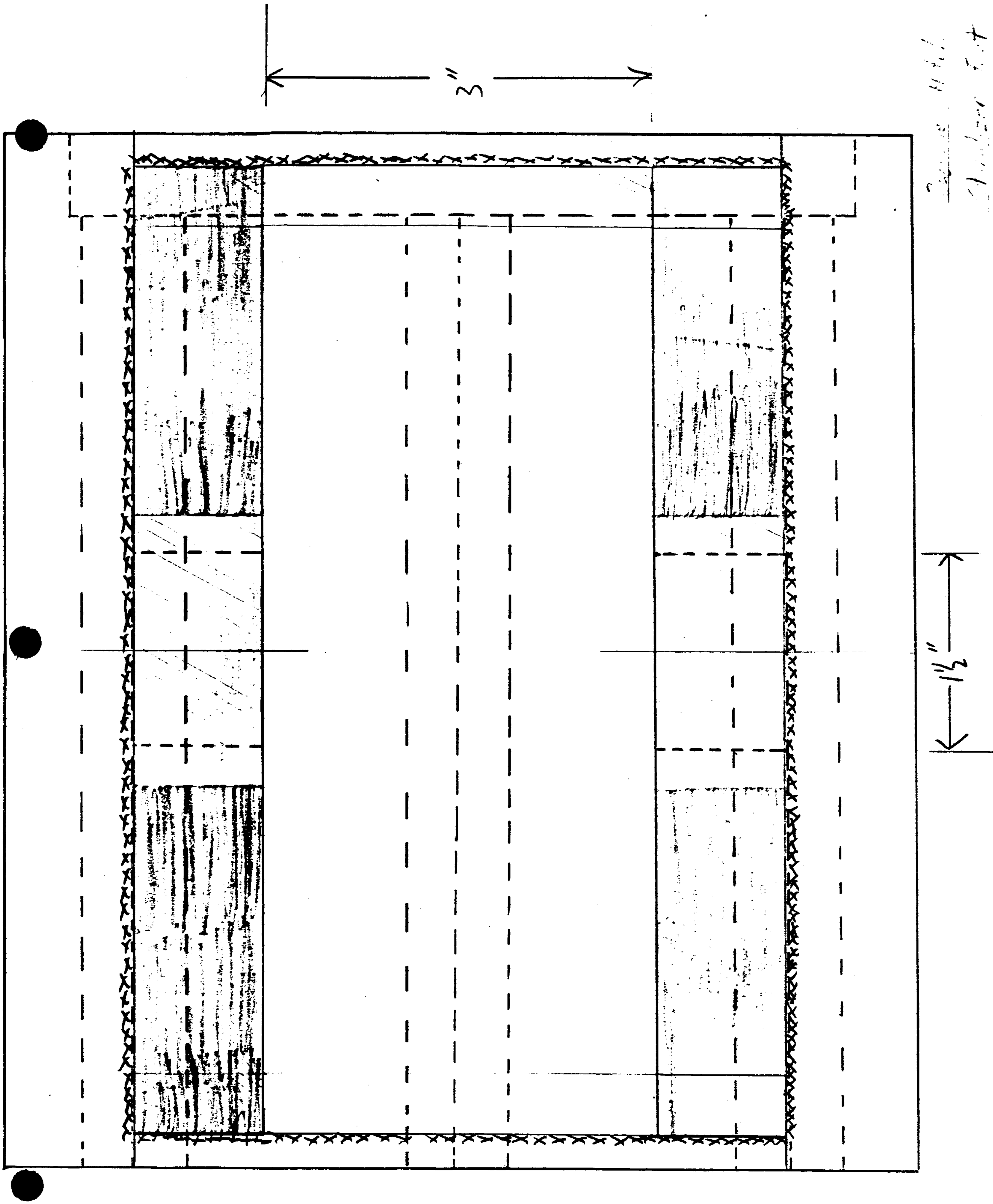
Backhoe Hitch
Rear View 1" = 4"
w/o C channel supports



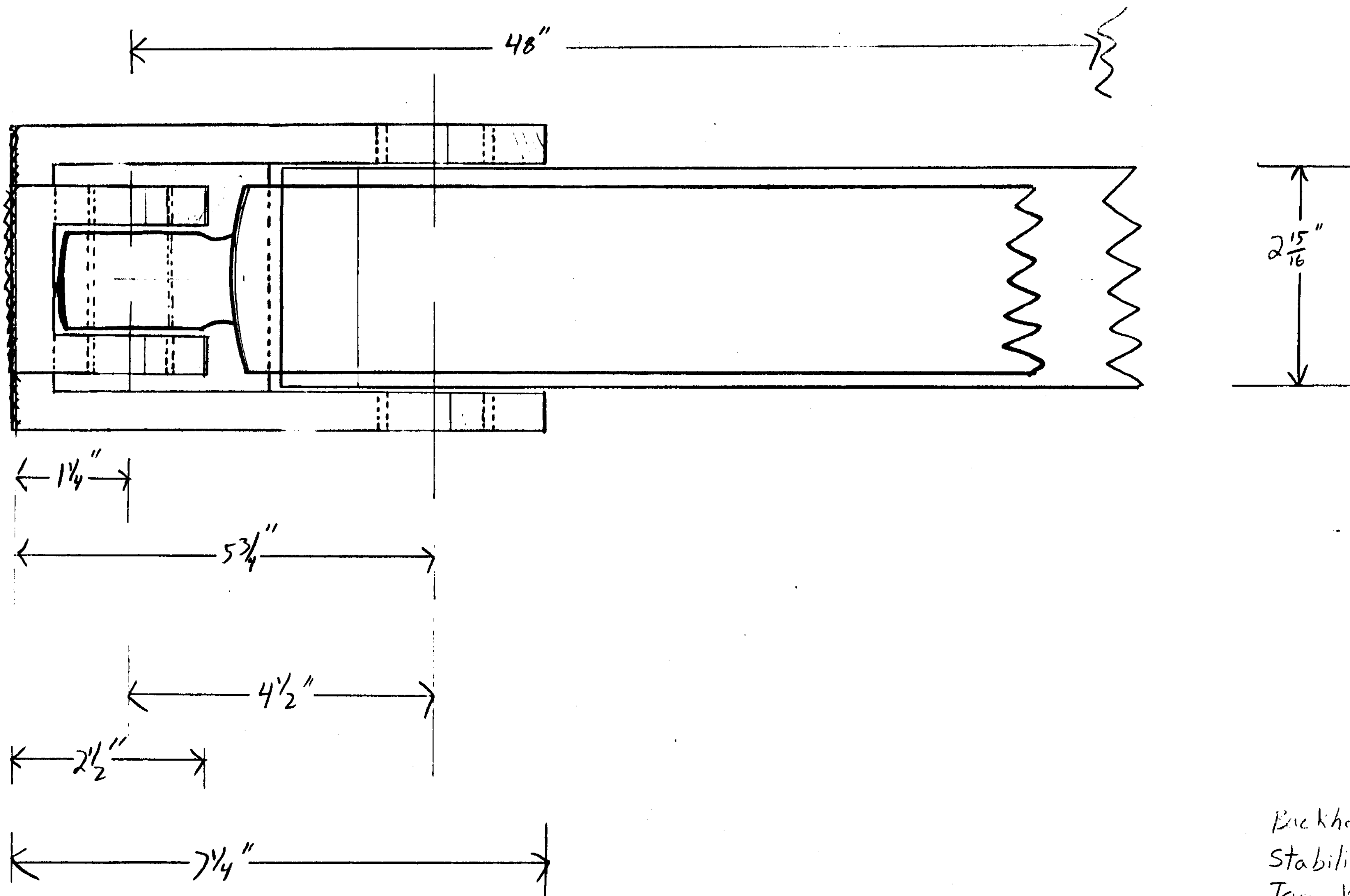
Backhoe Hitch
Stabilizer Foot
Front View 1"=1"



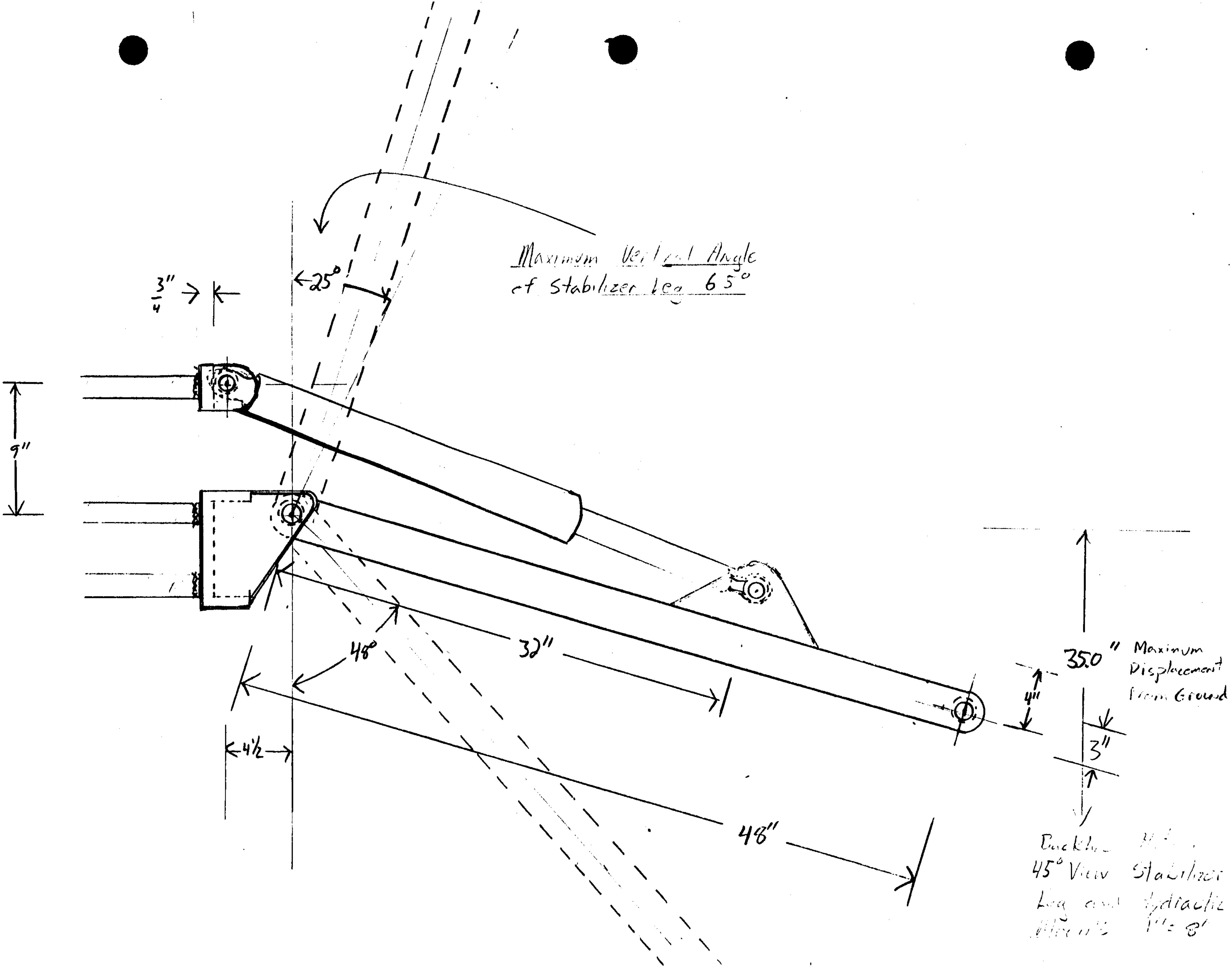
Backhoe Hitch
Stabilizer Foot
Side View
1:1



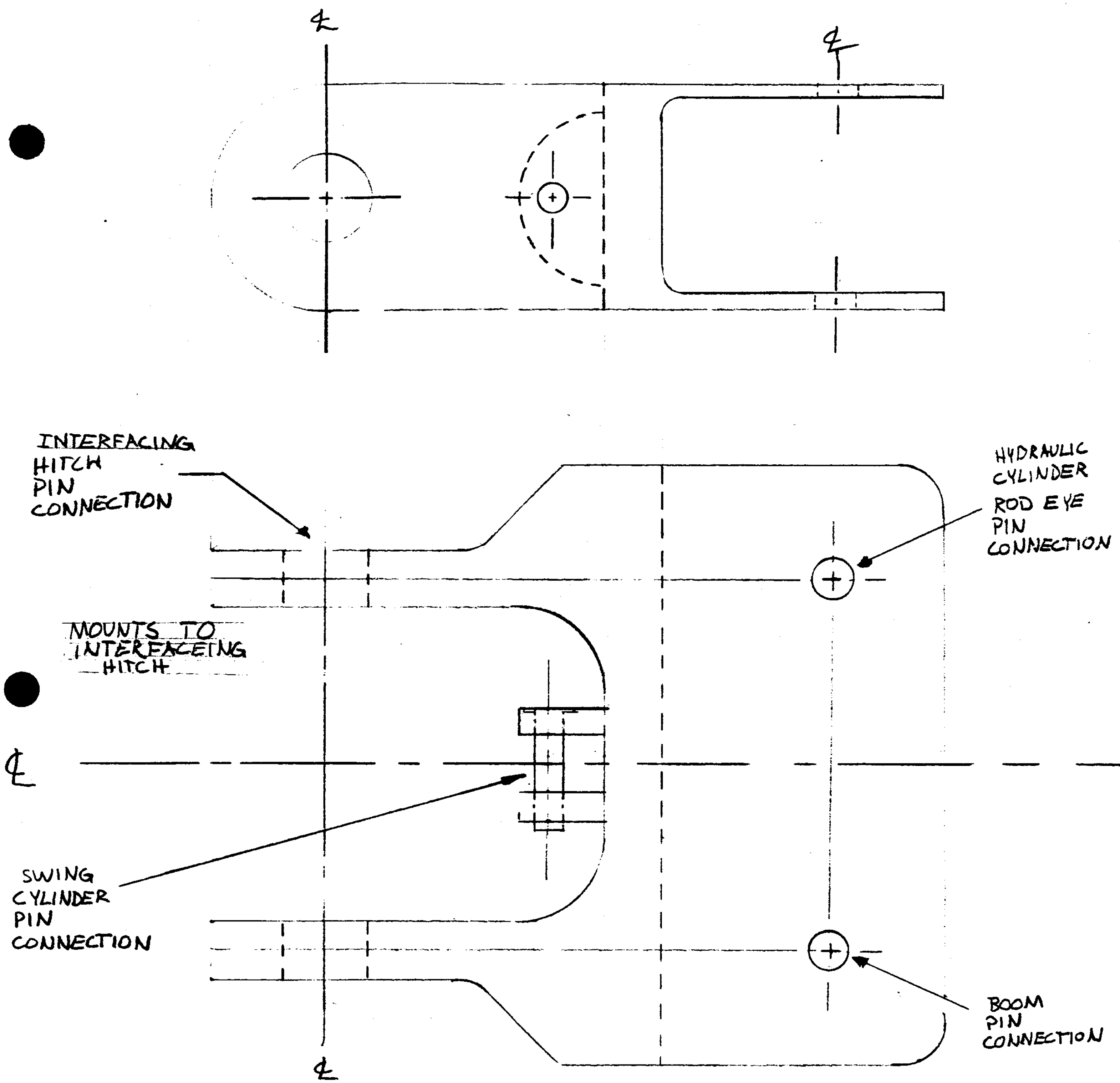
3"
1½"



Backhoe Hitch
Stabilizer Leg
Top View 1"=2"

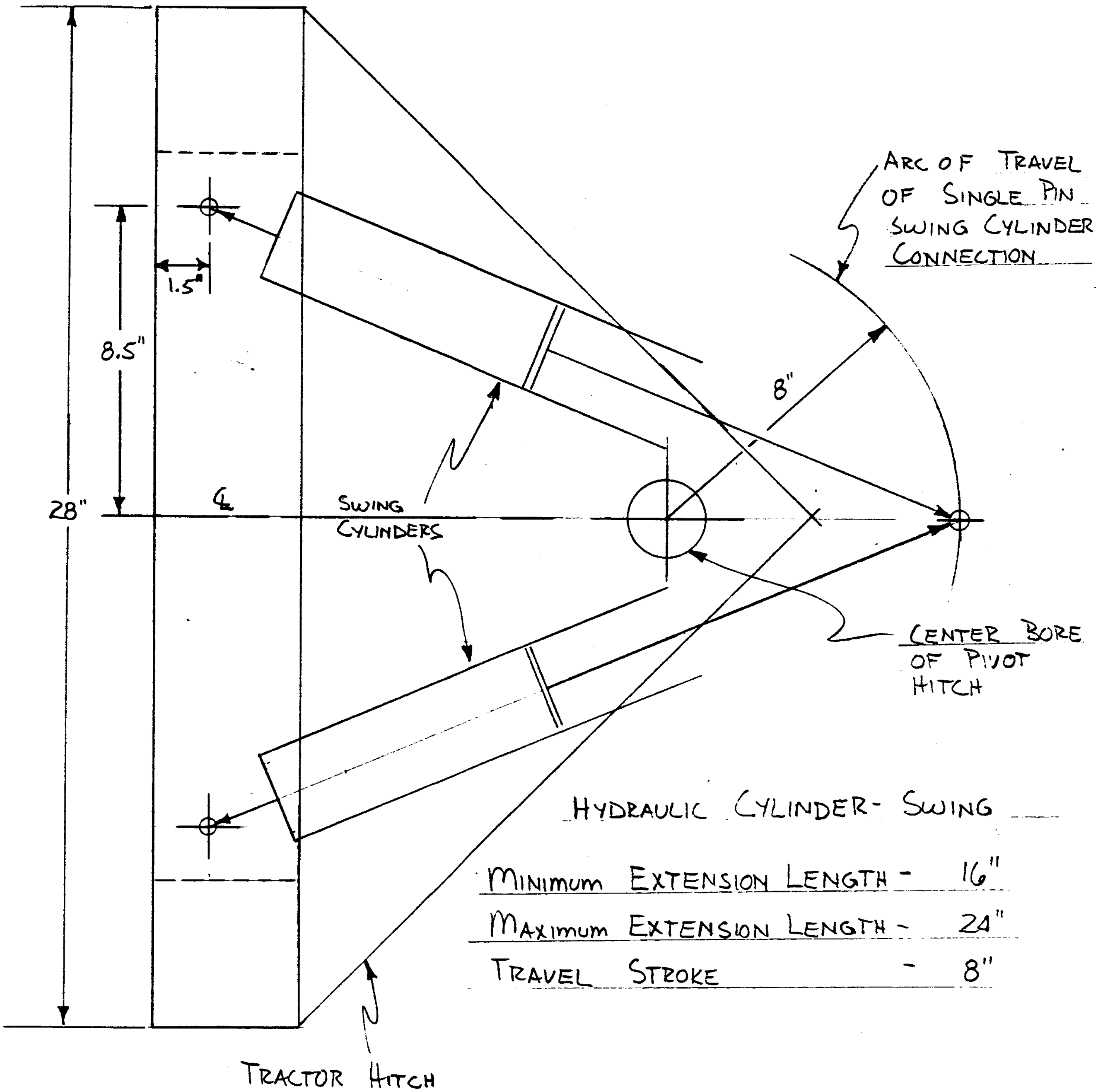


Maximum Vertical Angle of Stabilizer Leg 65°
25°
3/4"
9"
48°
32"
4½
48"
35.0" Maximum Displacement From Ground
4"
3"
45° View Stabilizer
Leg and Hydraulic



DETAILED/PIVOT HITCH

1/5" = 1"

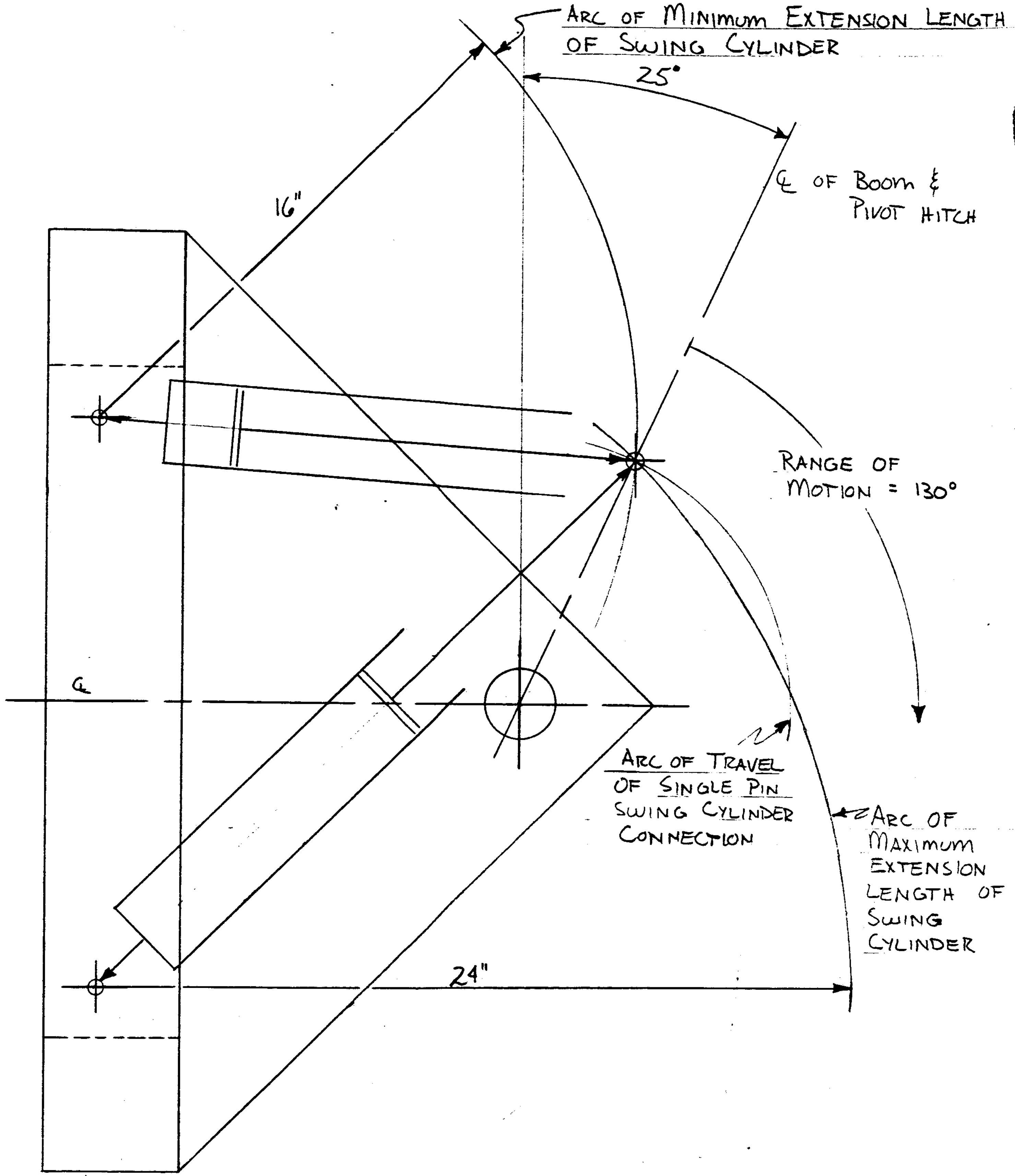


HYDRAULIC CYLINDER - SWING

MINIMUM EXTENSION LENGTH -	16"
MAXIMUM EXTENSION LENGTH -	24"
TRAVEL STROKE -	8"

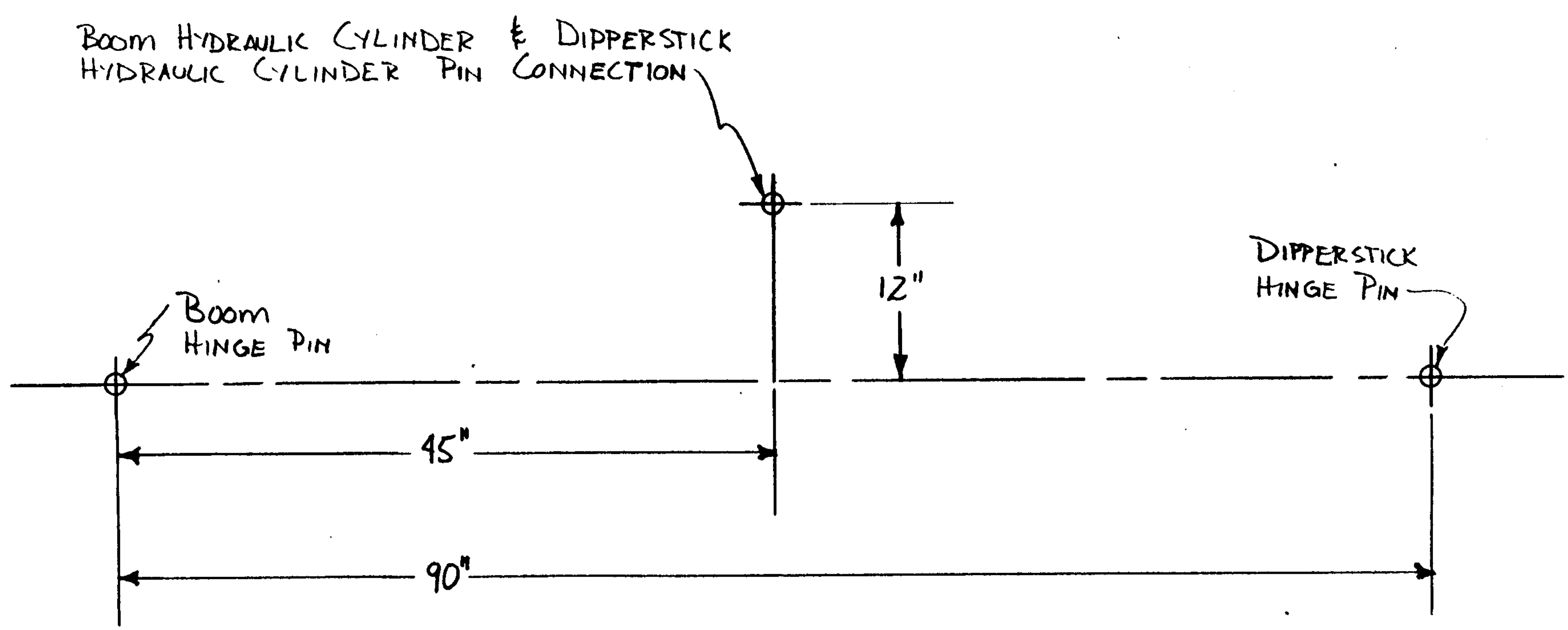
LOCATION & SPECIFICATIONS OF SWING CYLINDERS

SCALE : 1" = 4"



RANGE OF MOTION OF BOOM

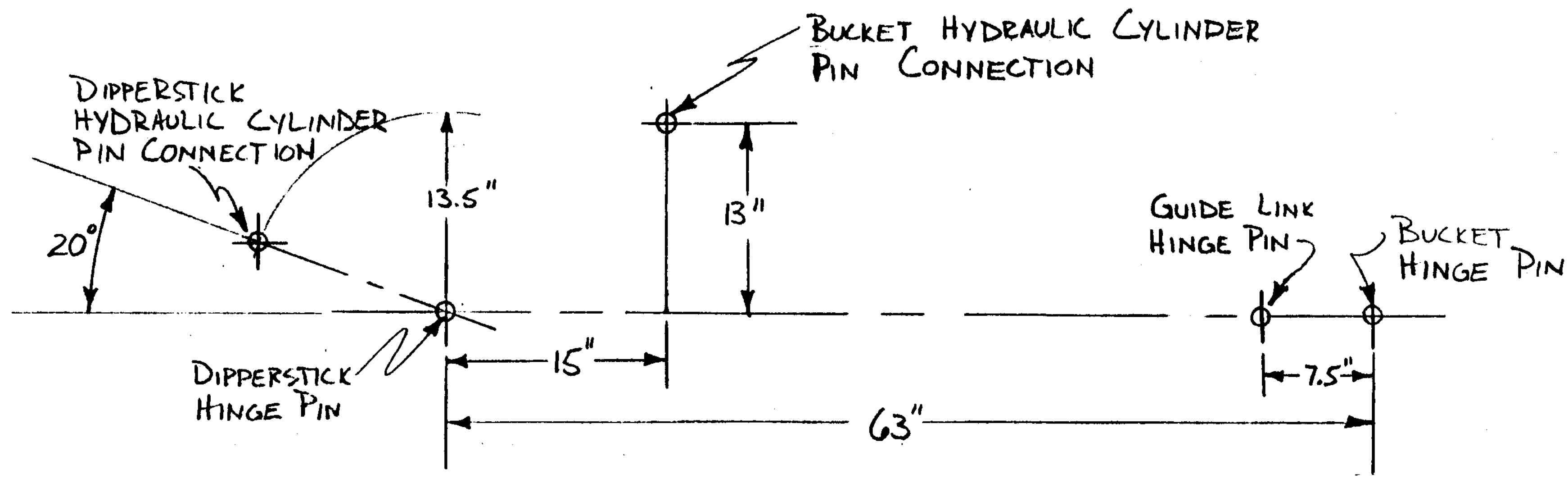
SCALE: 1" = 4"



HYDRAULIC CYLINDER - BOOM

MINIMUM EXTENSION LENGTH - 34"
MAXIMUM EXTENSION LENGTH - 58"
TRAVEL STROKE - 24"

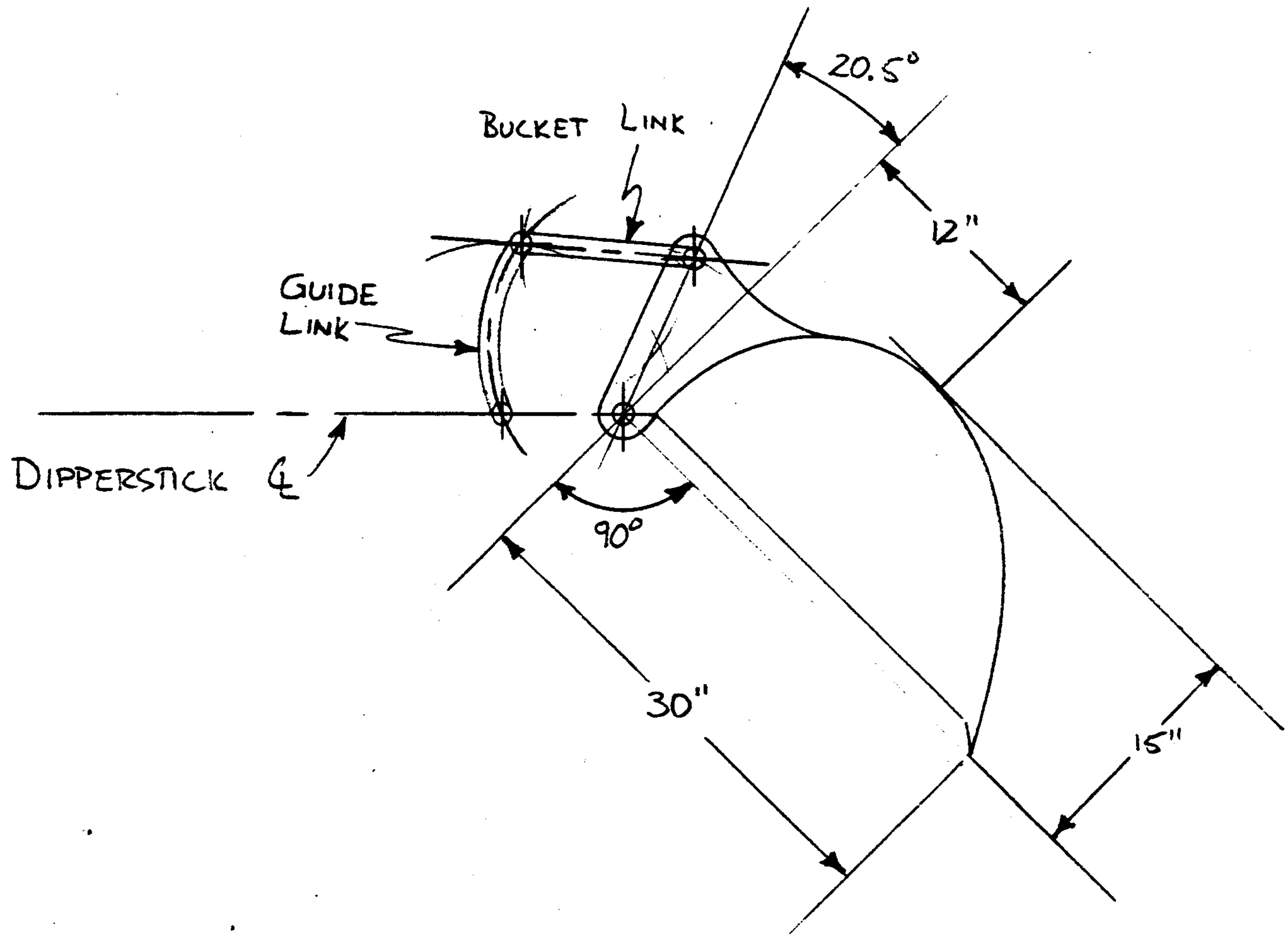
BOOM DIMENSIONS & BOOM
HYDRAULIC CYLINDER
SPECIFICATIONS
SCALE : 1" = 1'



HYDRAULIC CYLINDER - DIPPERSTICK

MINIMUM EXTENSION LENGTH - 34"
MAXIMUM EXTENSION LENGTH - 58"
TRAVEL STROKE - 24"

DIPPERSTICK DIMENSIONS AND DIPPERSTICK HYDRAULIC CYLINDER SPECIFICATIONS
SCALE : 1" = 1'

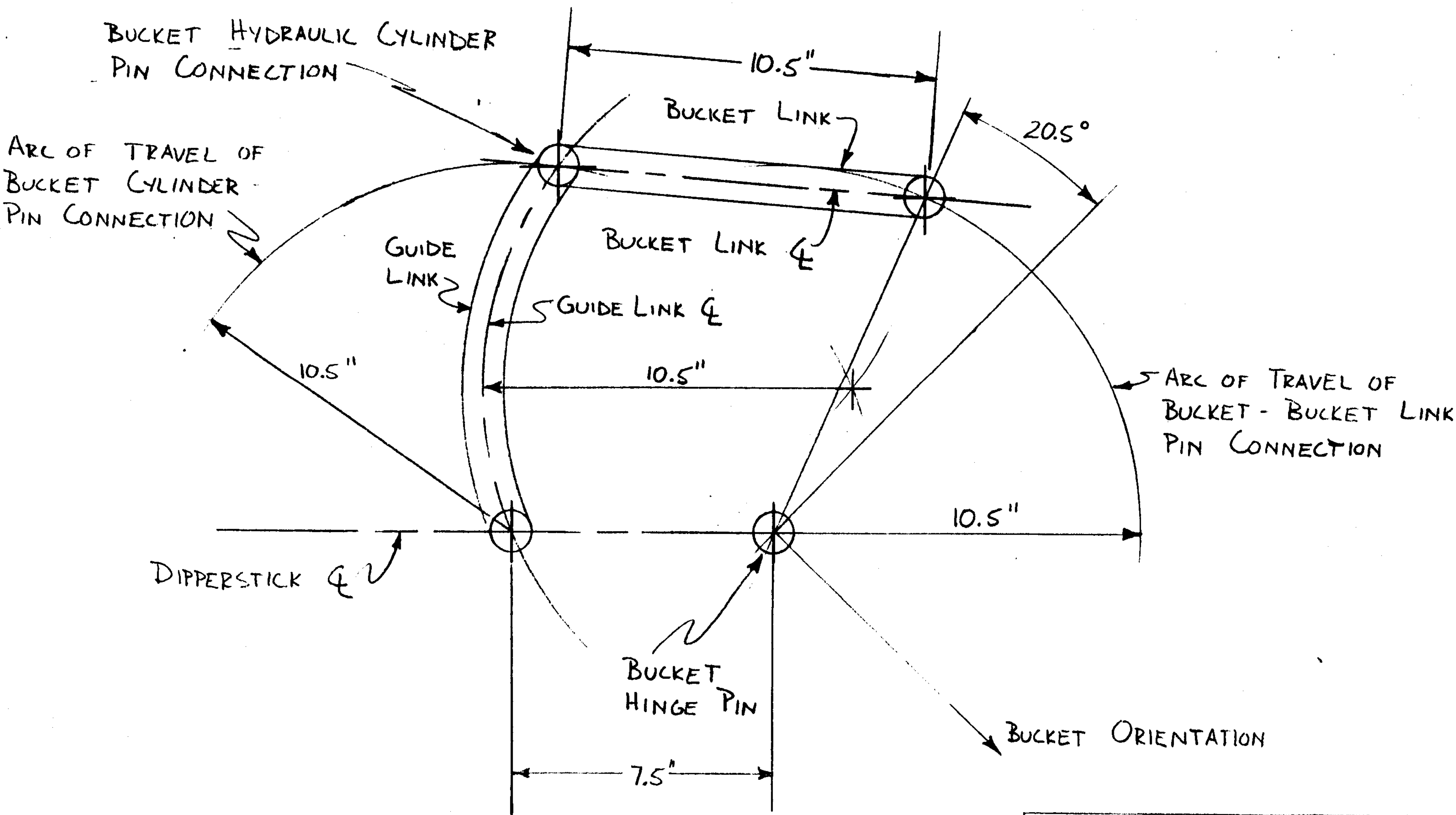


HYDRAULIC CYLINDER - BUCKET

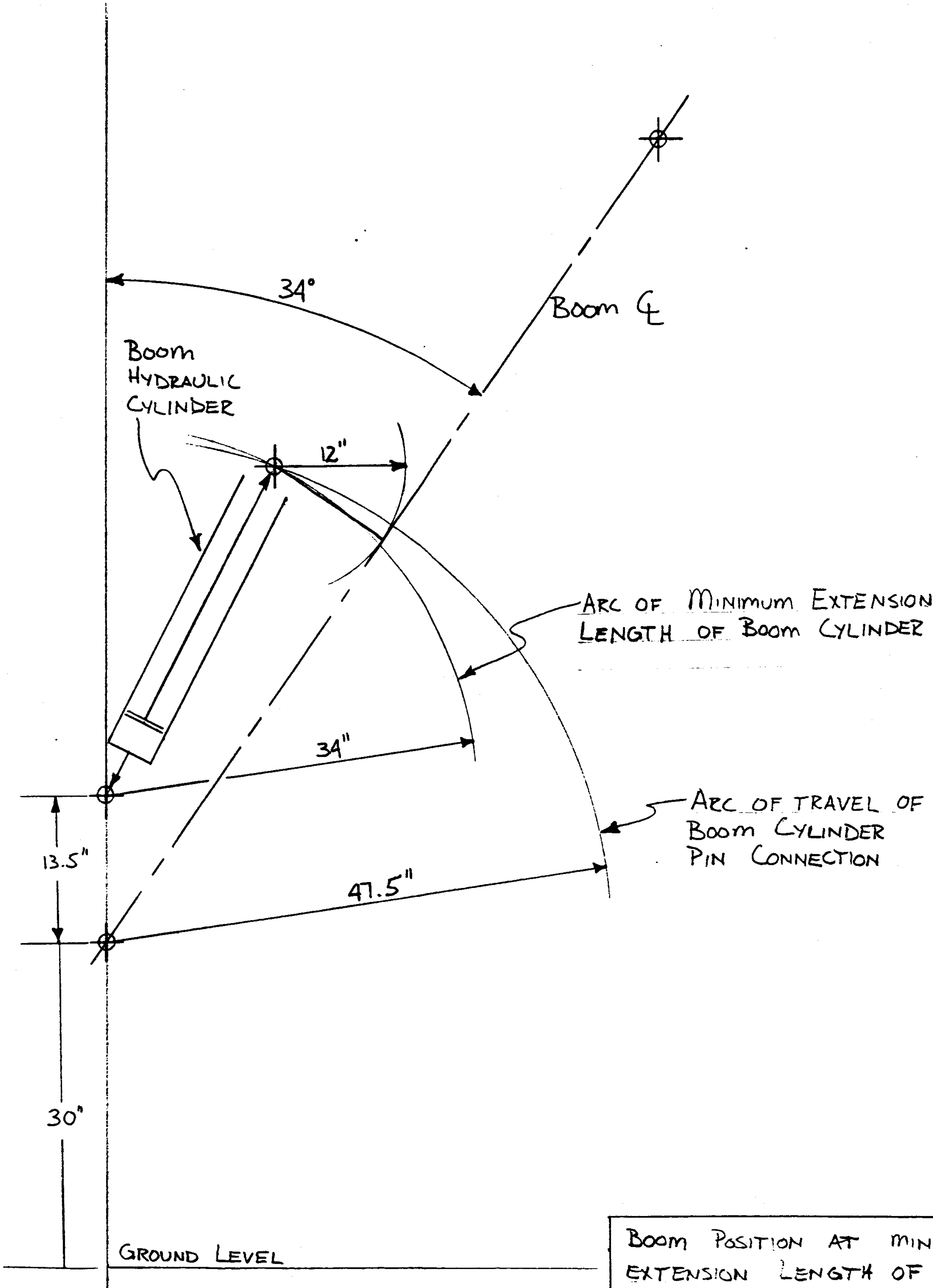
MINIMUM EXTENSION LENGTH - 34"
MAXIMUM EXTENSION LENGTH - 52"
TRAVEL STROKE - 18"

NOTE: SEE NEXT FIG. FOR ENLARGED VIEW OF BUCKET LINK & GUIDE LINK 4-BAR

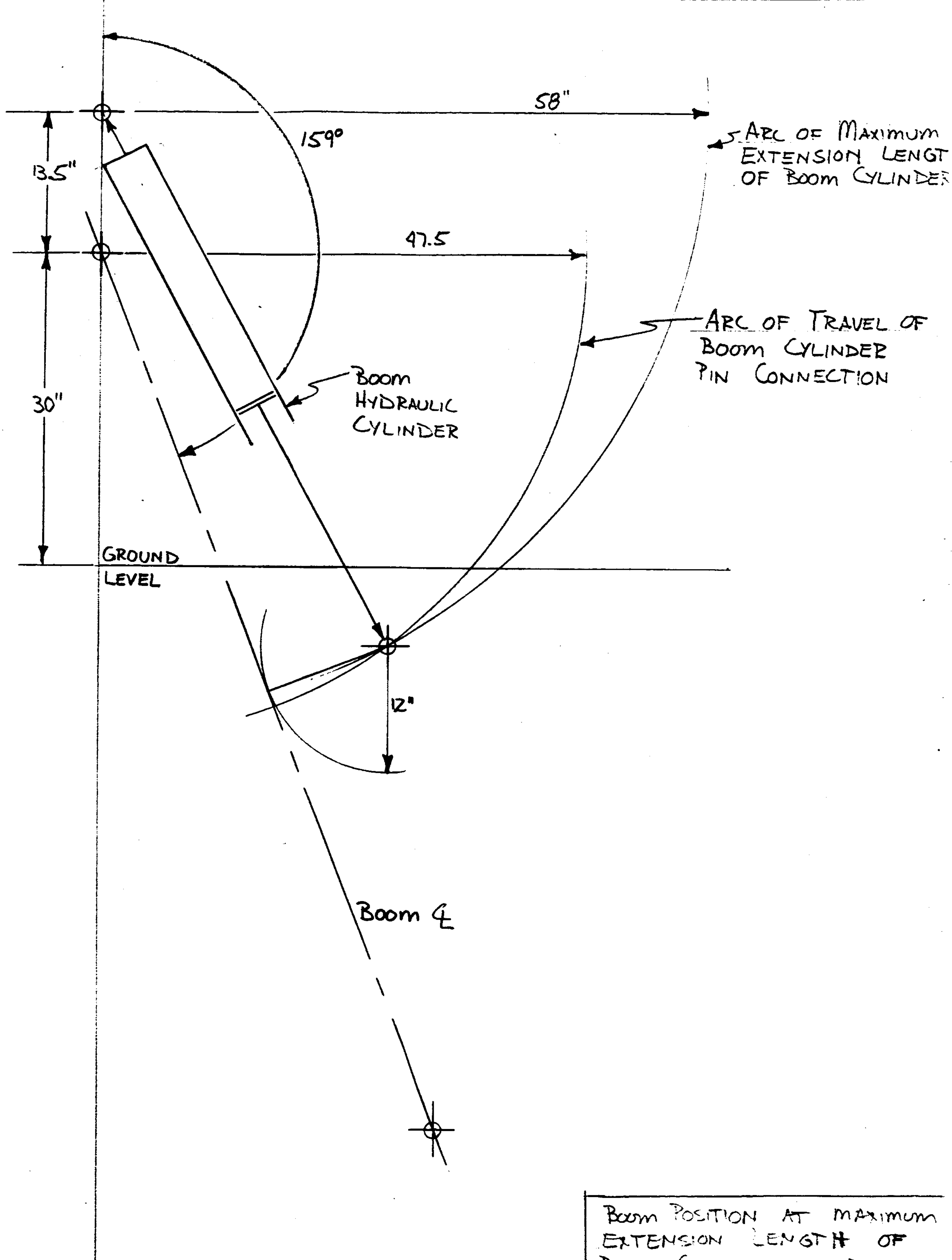
BUCKET AND BUCKET CYLINDER SPECIFICATIO[N]
SCALE : 1" = 1'



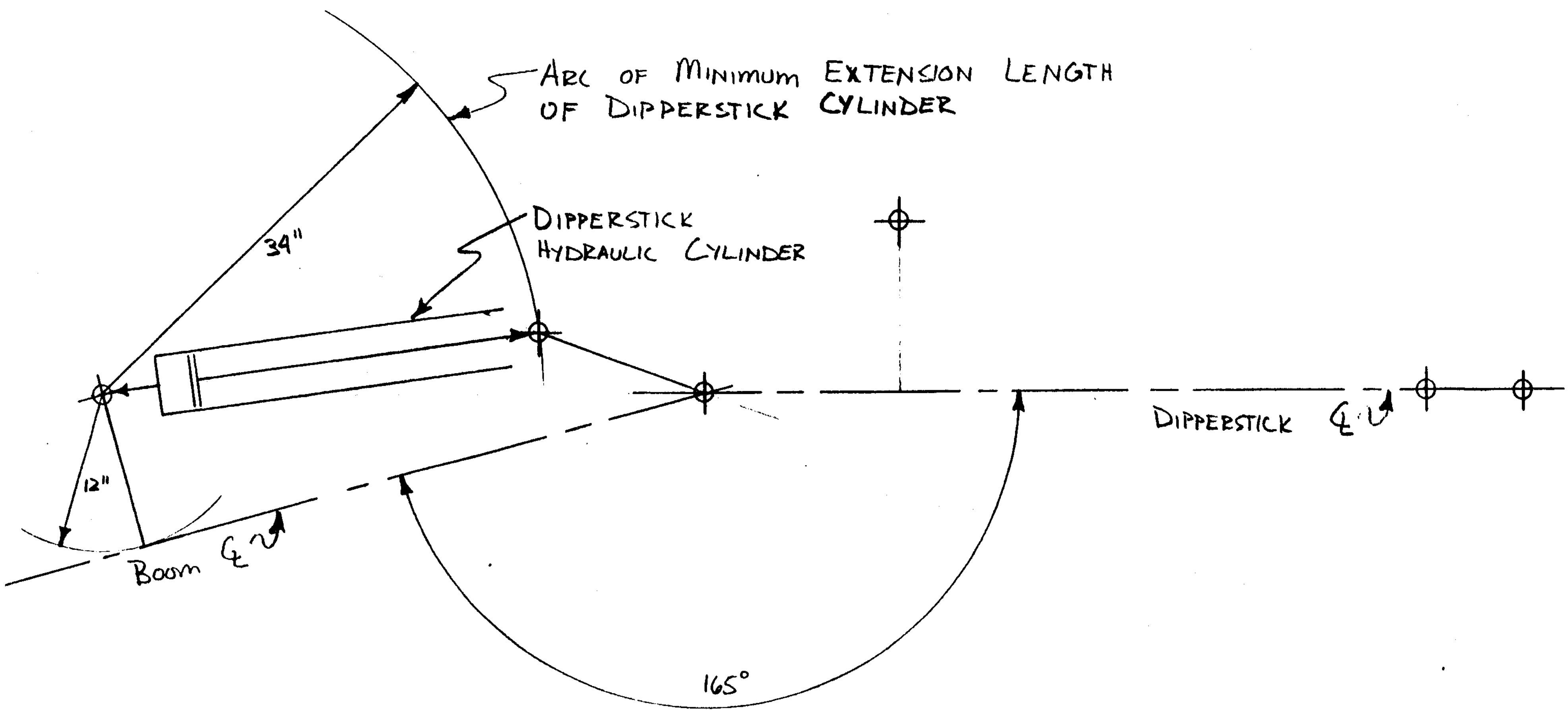
Guide Link, Bucket Link, Bucket & Dipperstick 4- Bar Dimensions
Scale : 1" = 4"



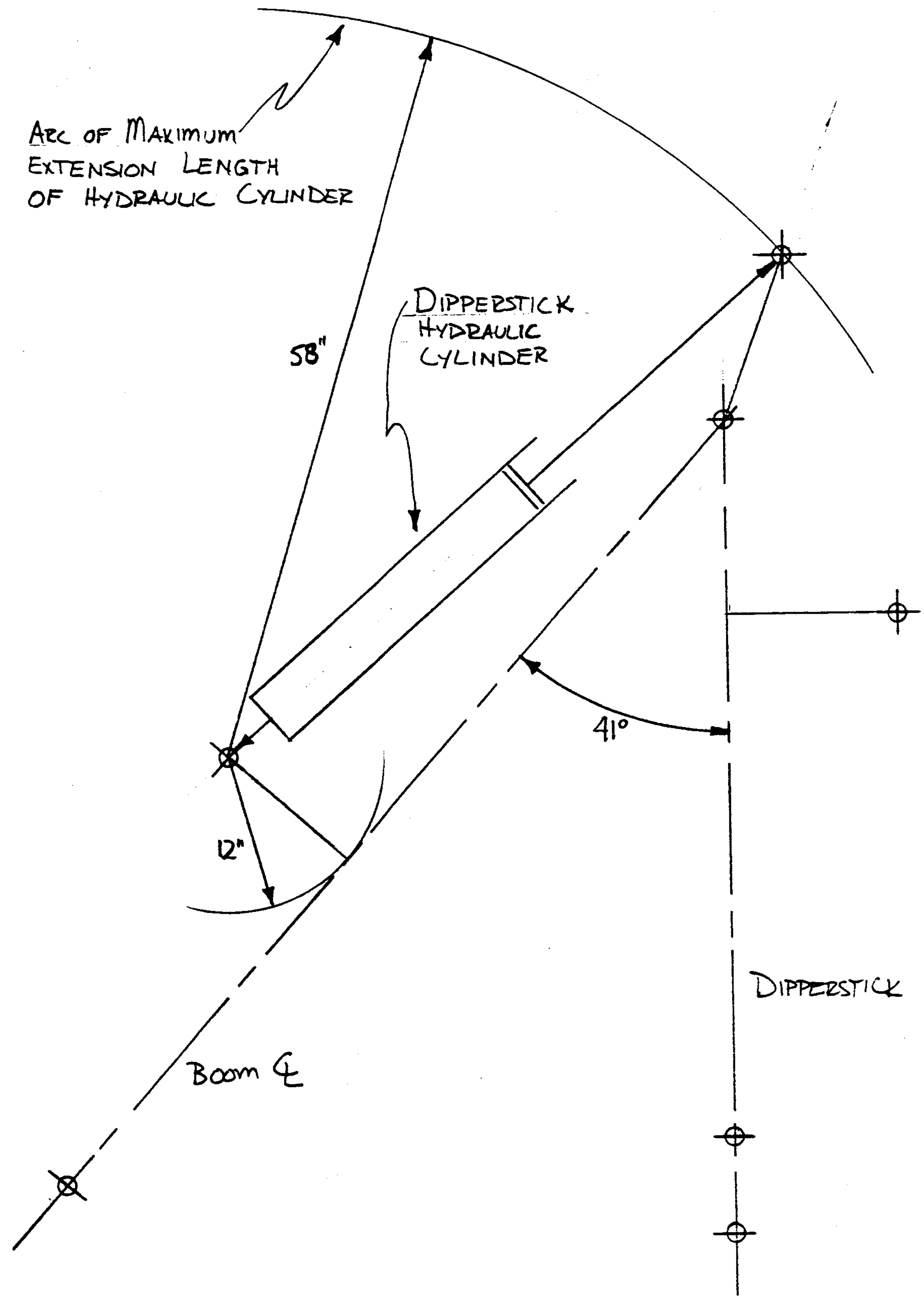
Boom Position at minimum
Extension Length of
Boom Cylinder WRT
Vertical Scale: 1"=



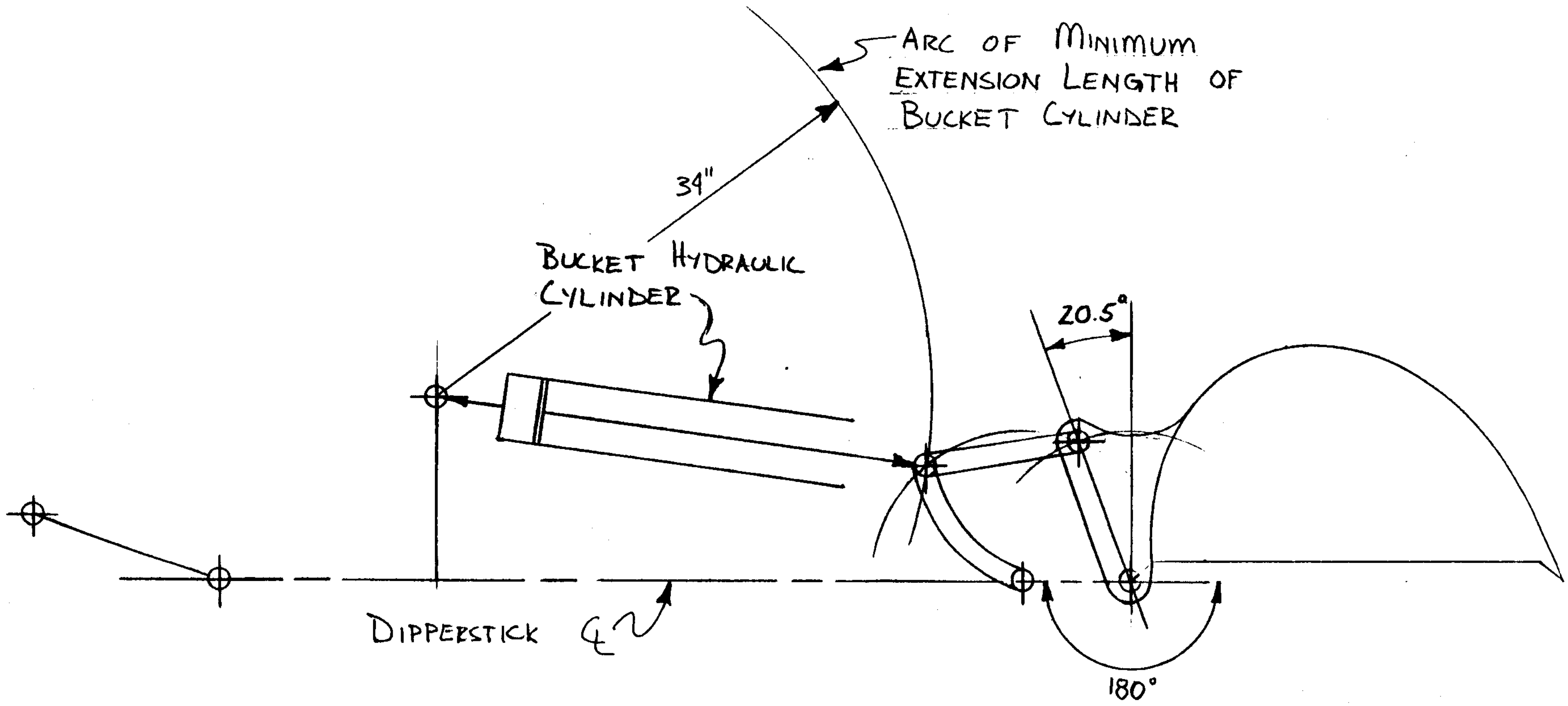
BOOM POSITION AT MAXIMUM EXTENSION LENGTH OF BOOM CYLINDER WRT VERTICAL SCALE: 1"=1'



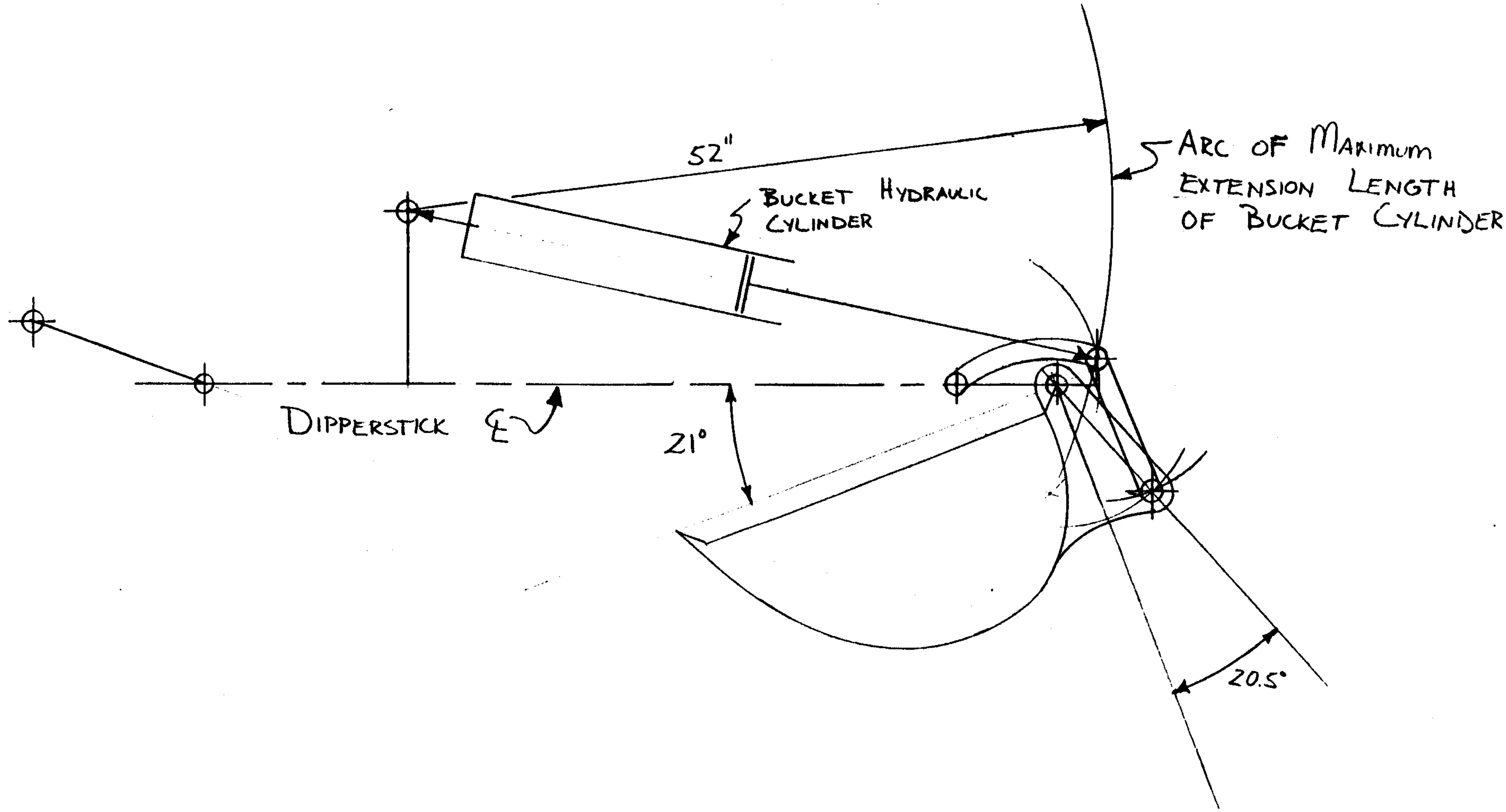
DIPPERSTICK POSITION AT
MINIMUM EXTENSION LENTGH
OF CYLINDER WRT BOOM
SCALE : 1" = 1'



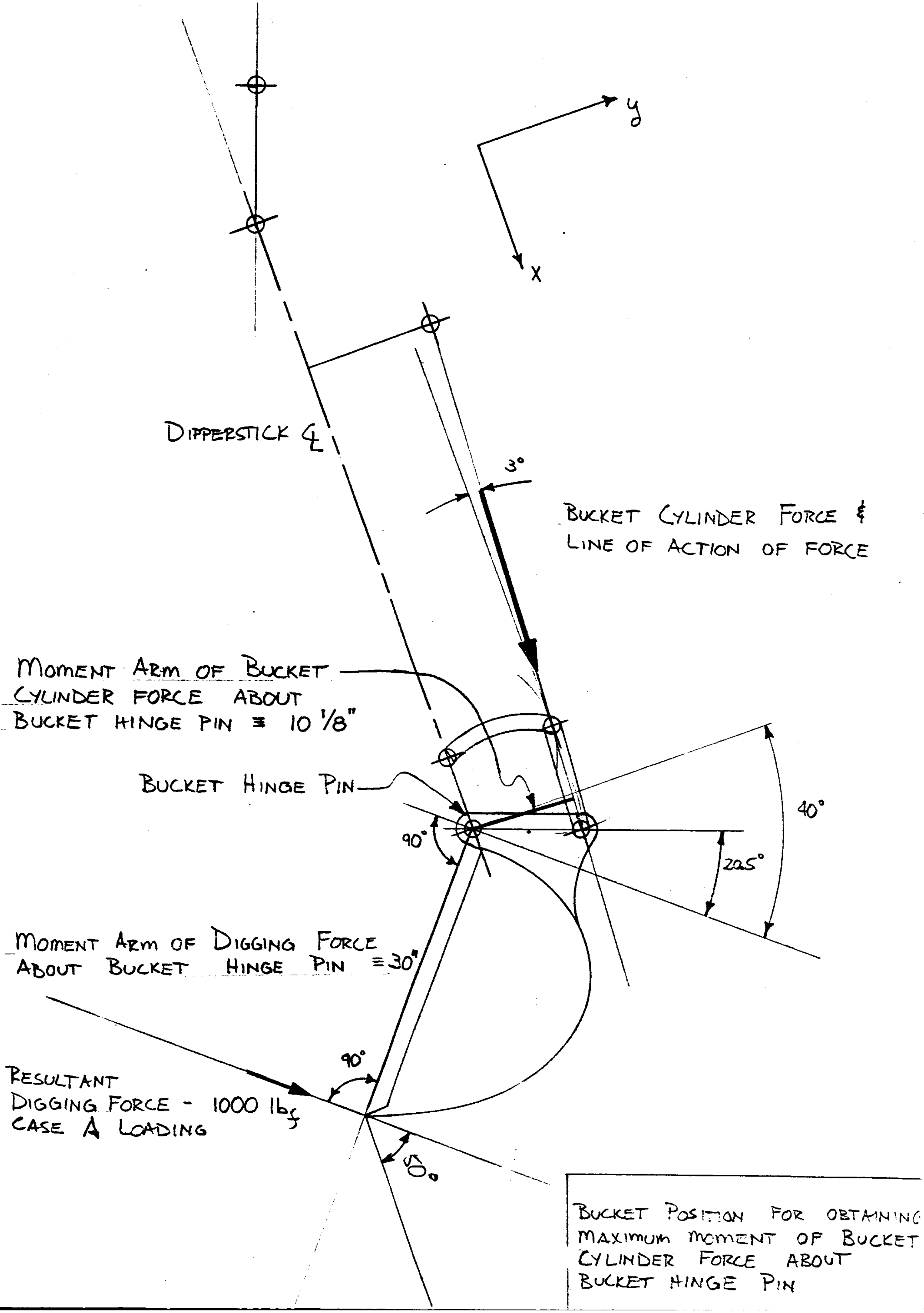
DIPPERSTICK POSITION AT
MAXIMUM EXTENSION LENGTH
OF CYLINDER WRT BOOM
SCALE : 1" = 1'



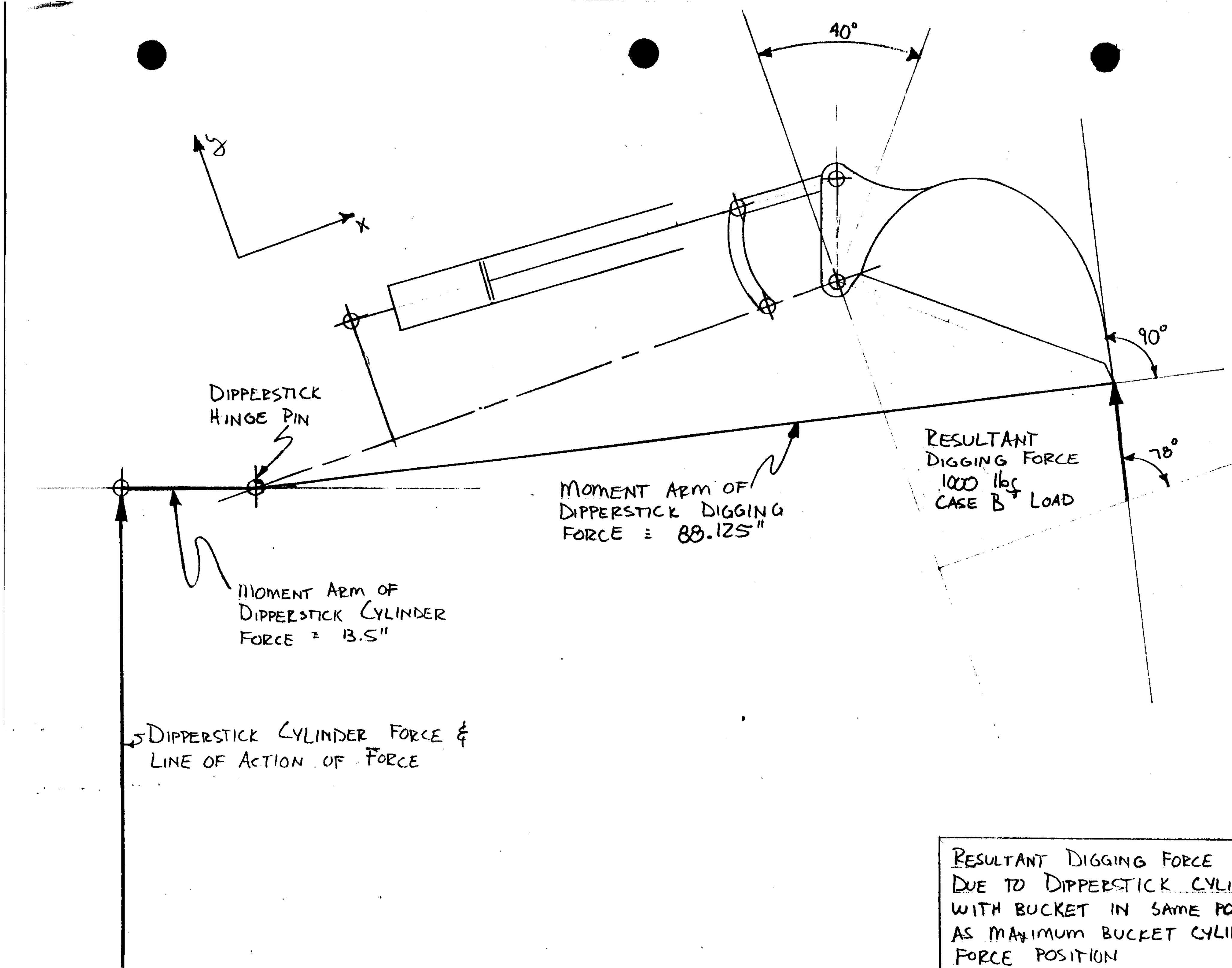
BUCKET POSITION WITH RESPECT TO DIPPERSTICK ℄ : BUCKET CYLINDER AT MINIMUM EXTENSION LENGTH



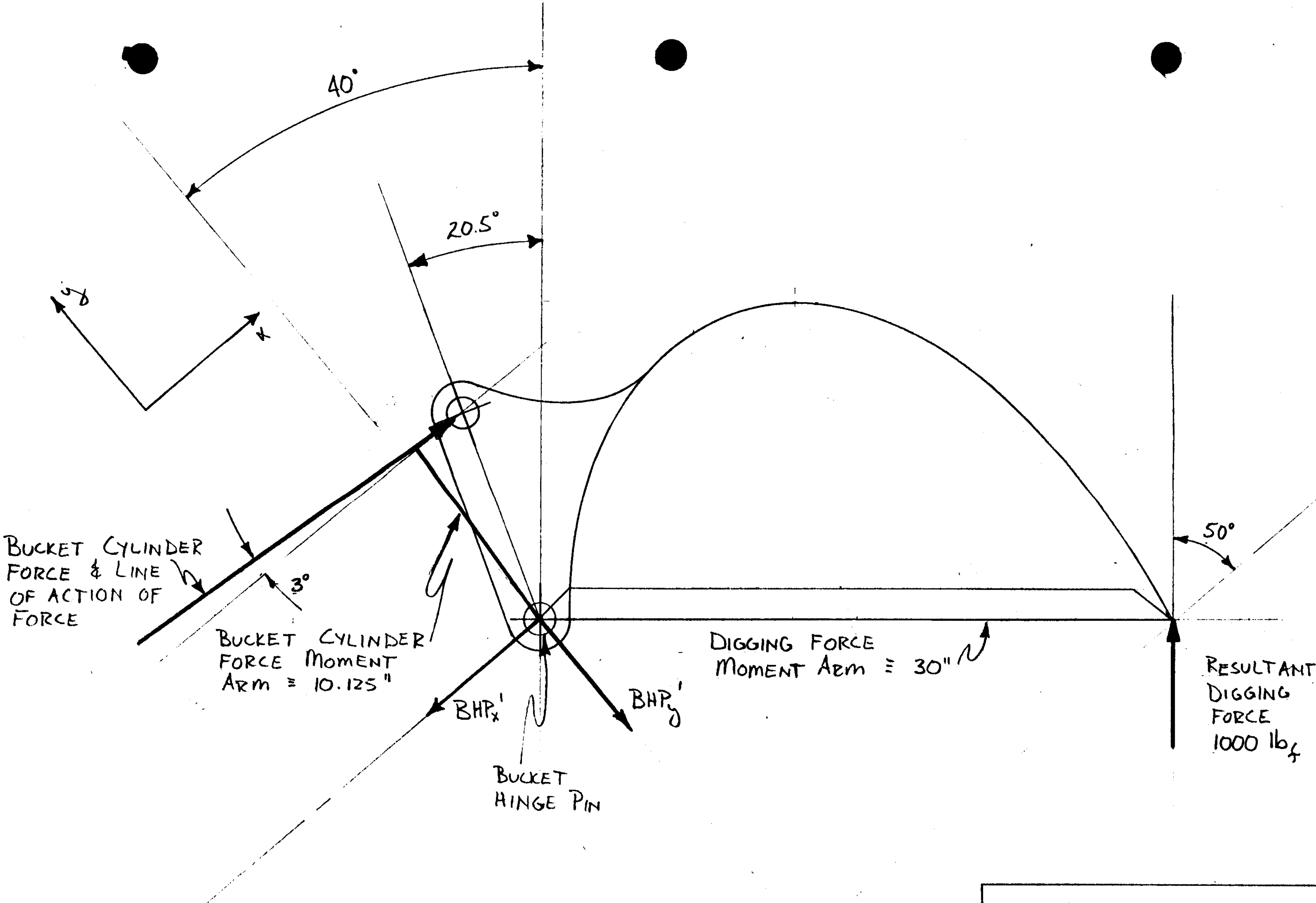
BUCKET POSITION WITH RESPECT TO DIPPERSTICK ℄:
BUCKET POSITION AT MAXIMUM EXTENSION LENGTH
SCALE : 1" = 1'



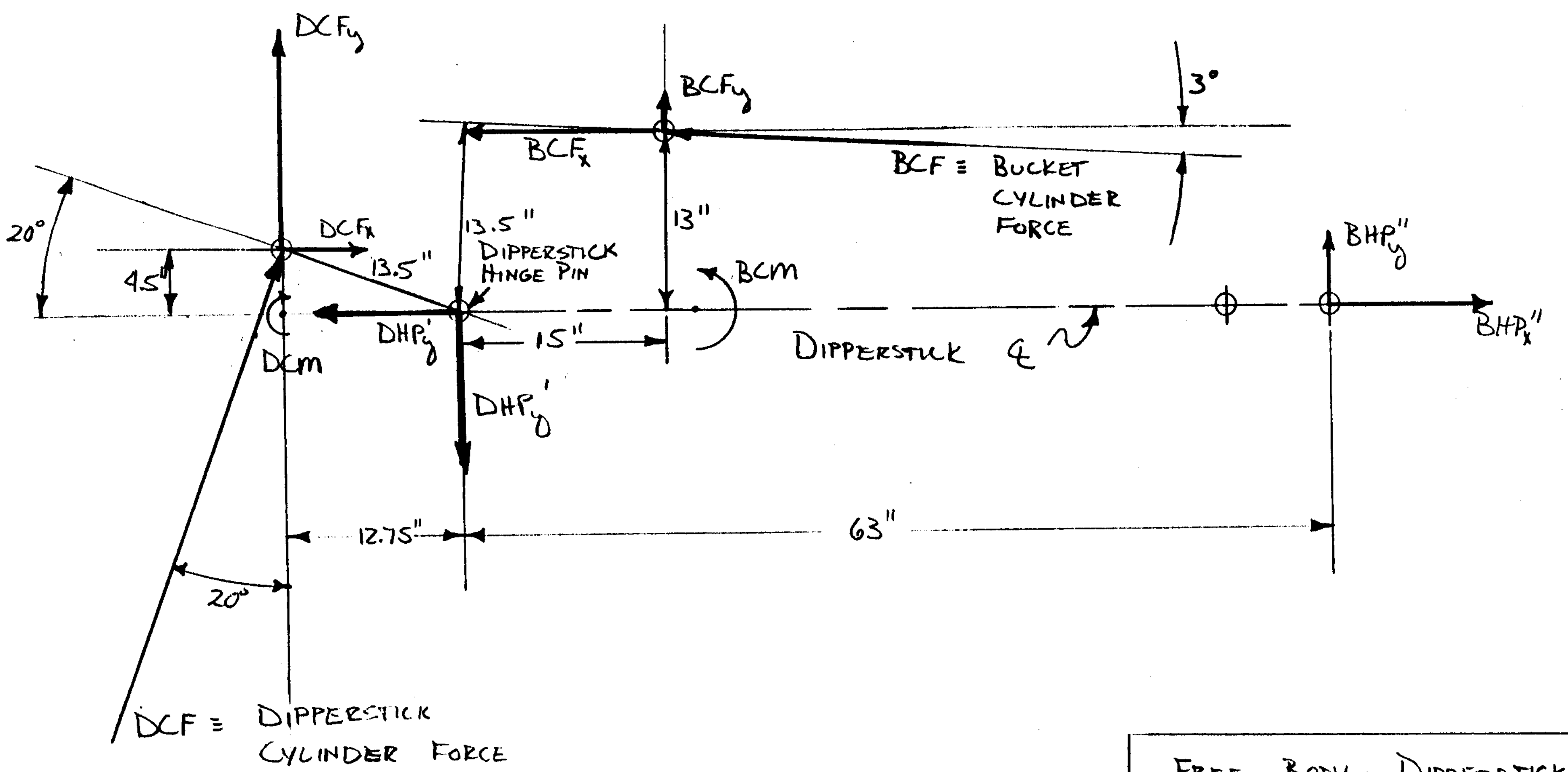
BUCKET POSITION FOR OBTAINING MAXIMUM MOMENT OF BUCKET CYLINDER FORCE ABOUT BUCKET HINGE PIN



RESULTANT DIGGING FORCE
DUE TO DIPPERSTICK CYLINDE[R]
WITH BUCKET IN SAME POSITI[ON]
AS MAXIMUM BUCKET CYLINDE[R]
FORCE POSITION

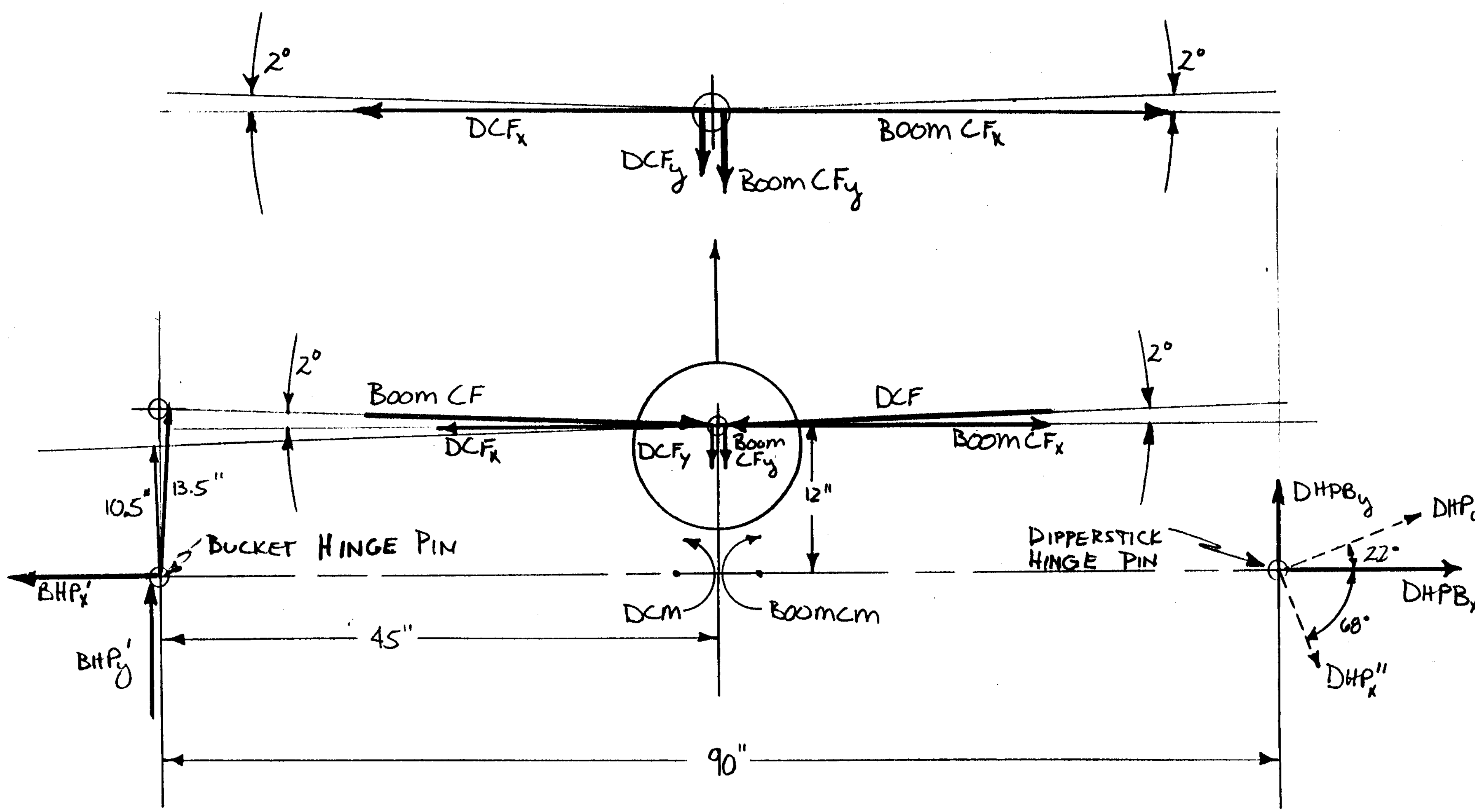


FREE BODY - BUCKET
SCALE : 1" = 6"



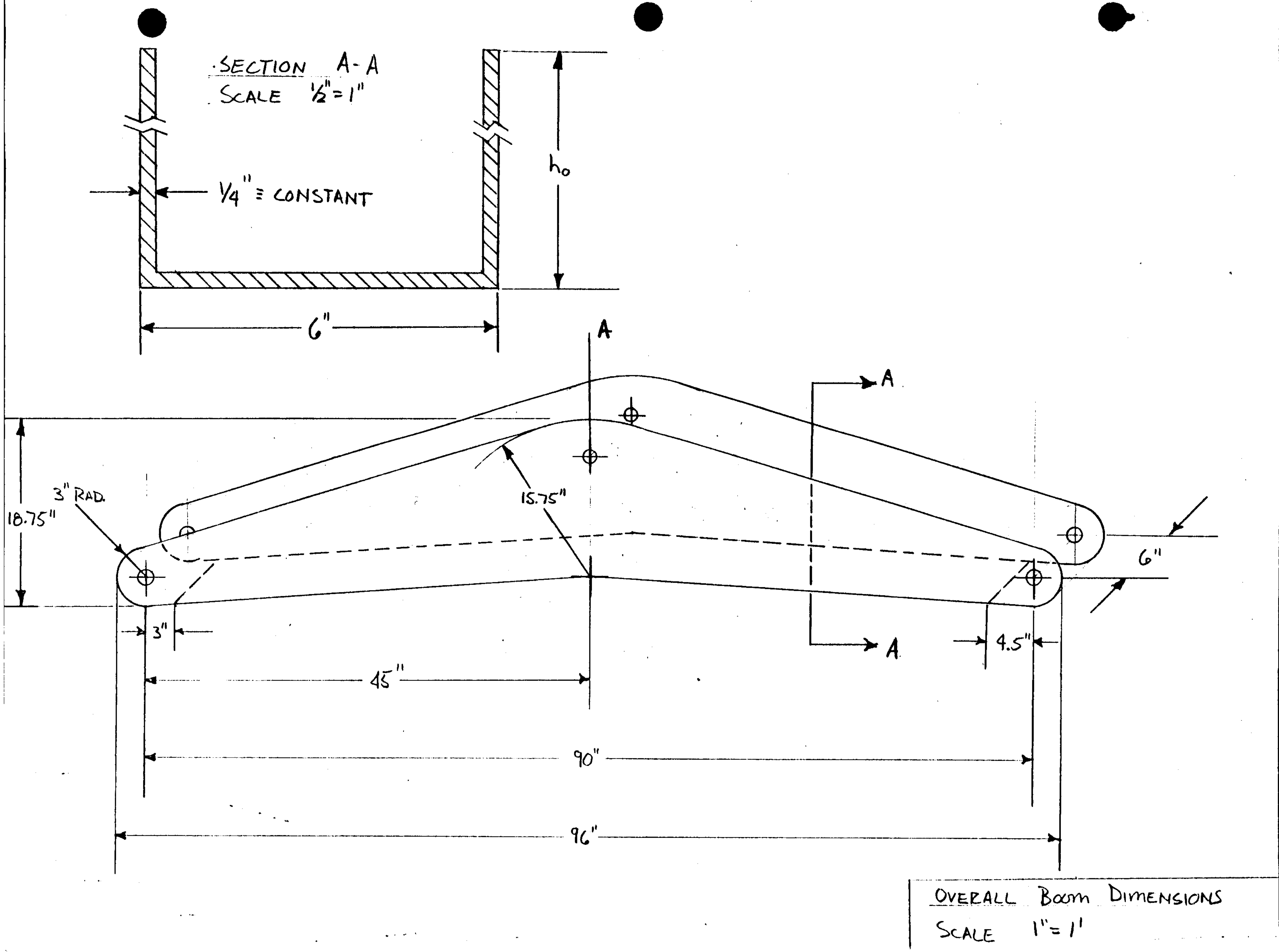
FREE BODY - DIPPERSTICK

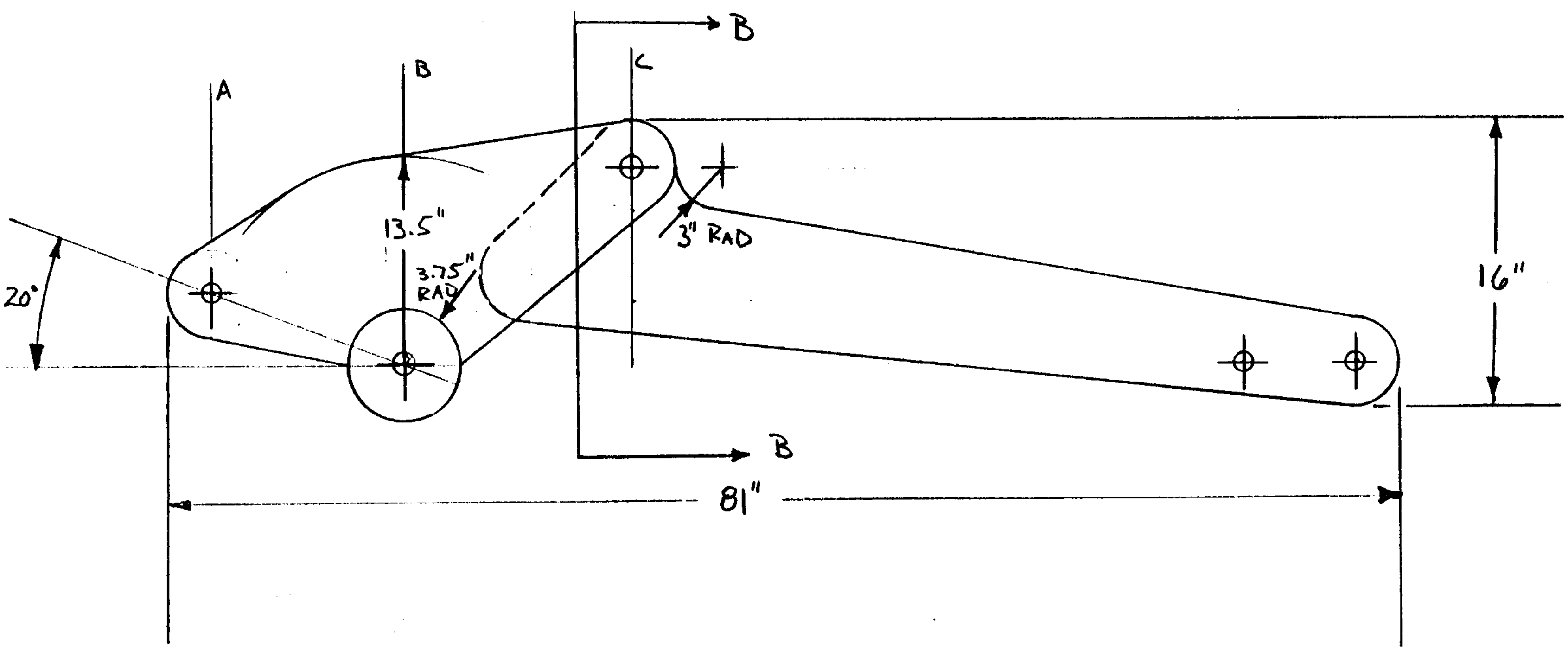
SCALE : 1" = 1'



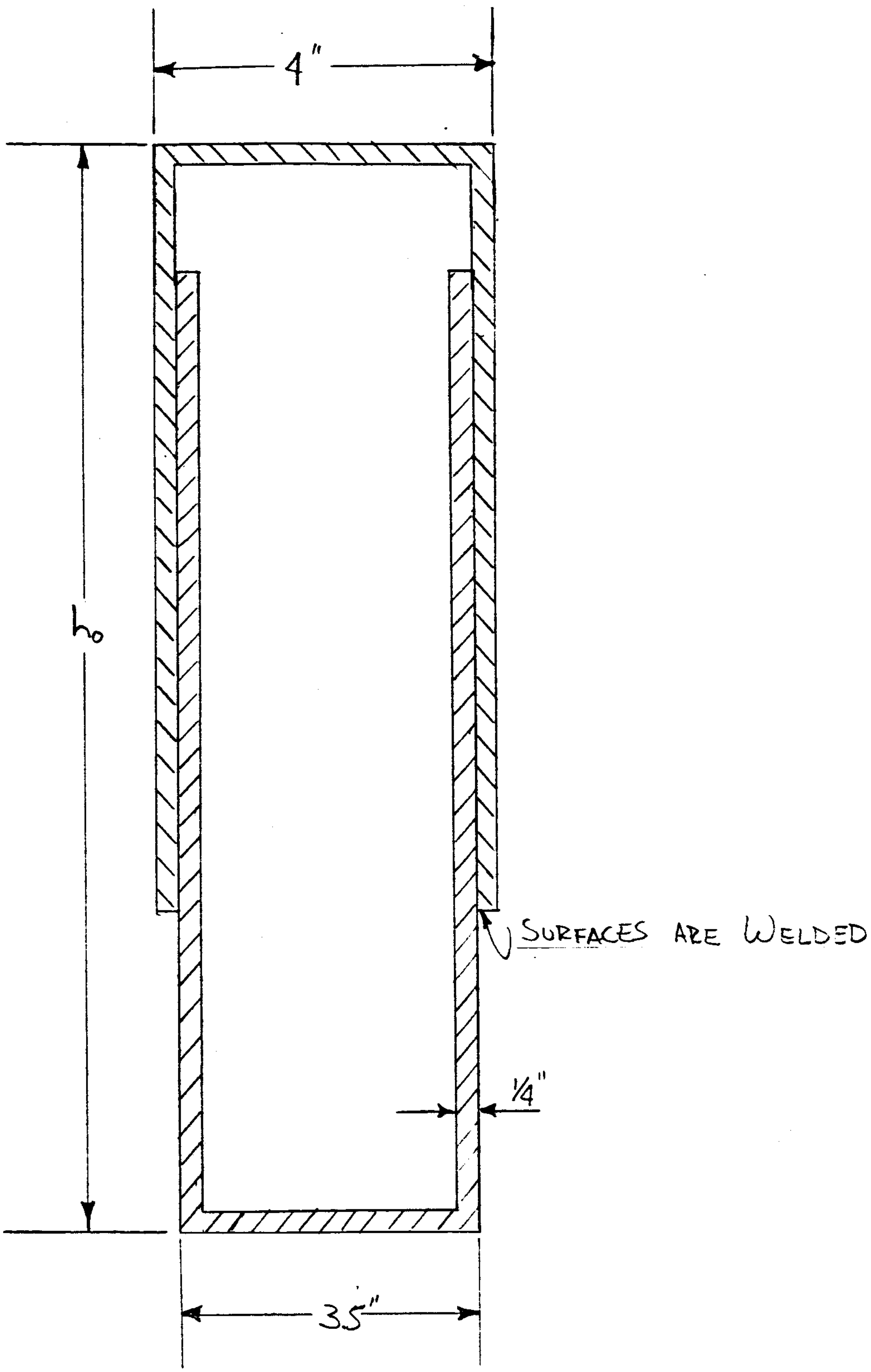
FREE BODY - BOOM

SCALE : 1" = 1'


SECTION A-A
SCALE 1/2" = 1"
1/4" ≡ CONSTANT
h_o
6"
A
A
A
3" RAD.
18.75"
15.75"
6"
3"
45"
4.5"
90"
96"
OVERALL BOOM DIMENSIONS
SCALE 1" = 1'



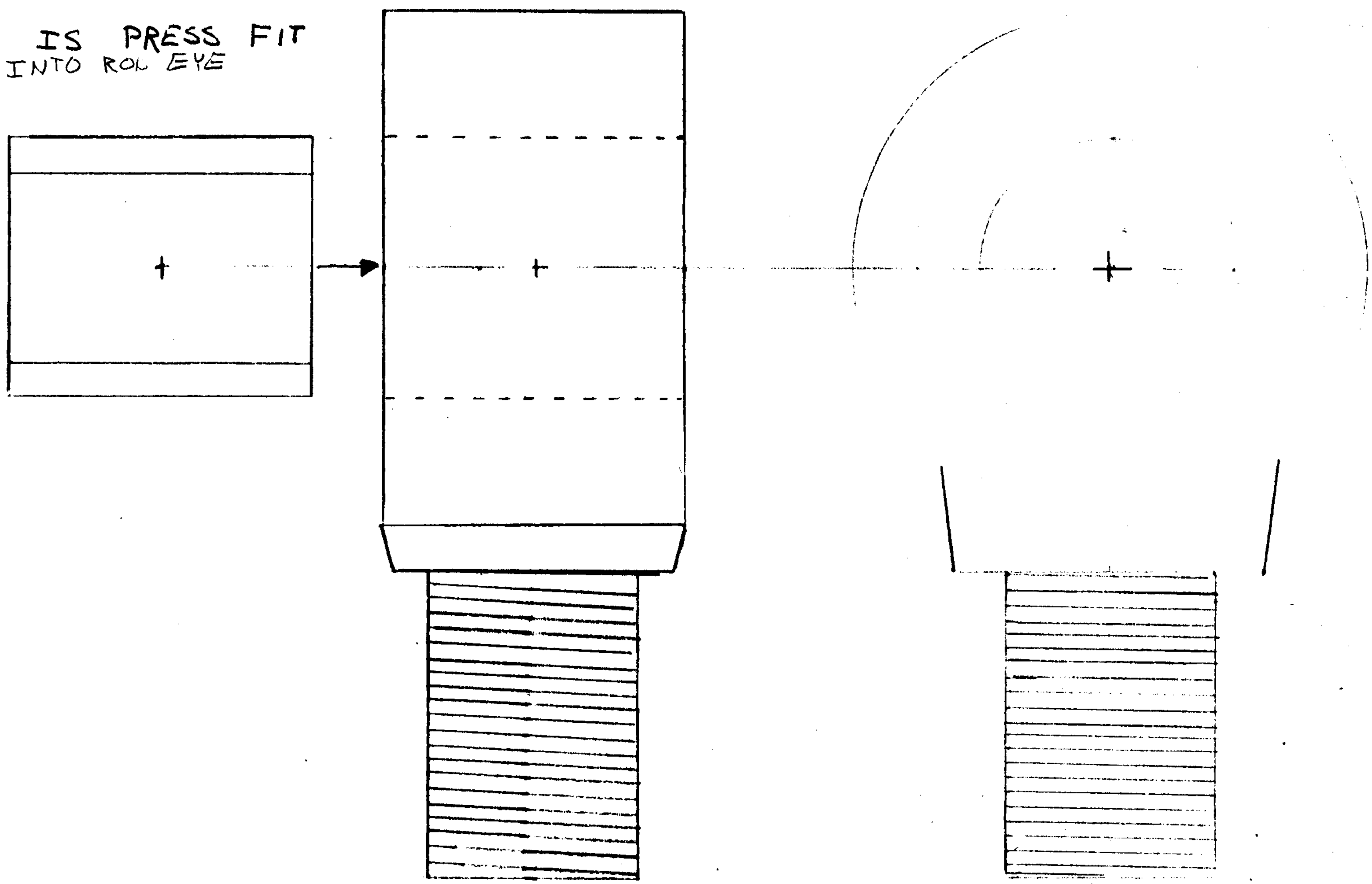
OVERALL DIPPERSTICK DIMENSIONS
SCALE : 1" = 1'



DETAIL OF WELDED SECTION OF DIPPERSTICK

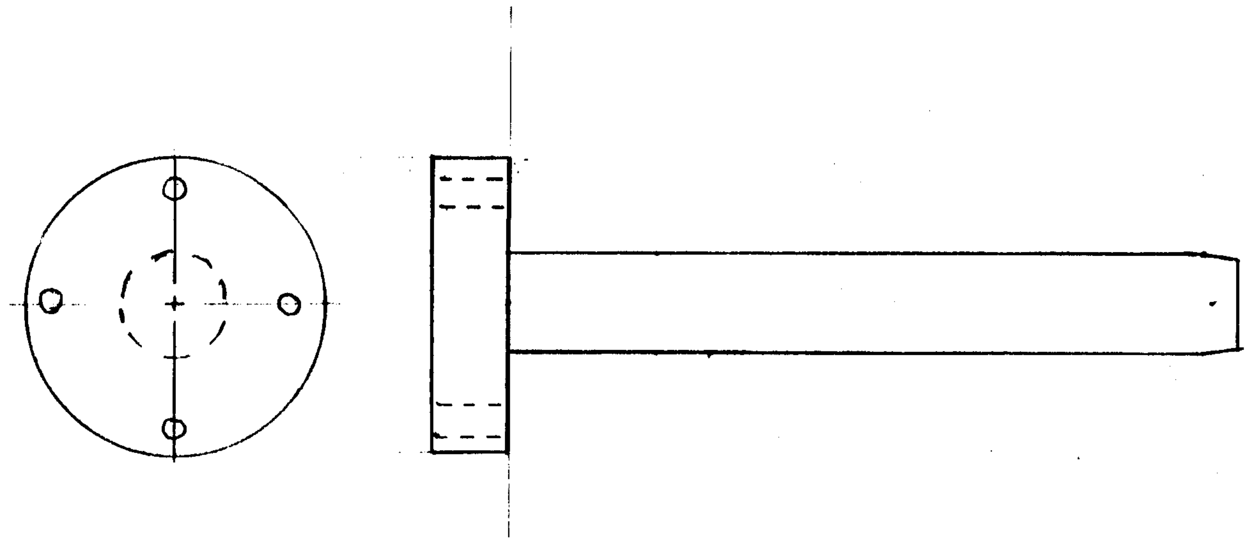
SCALE: 1" = ½"

BUSHING IS PRESS FIT INTO ROD EYE



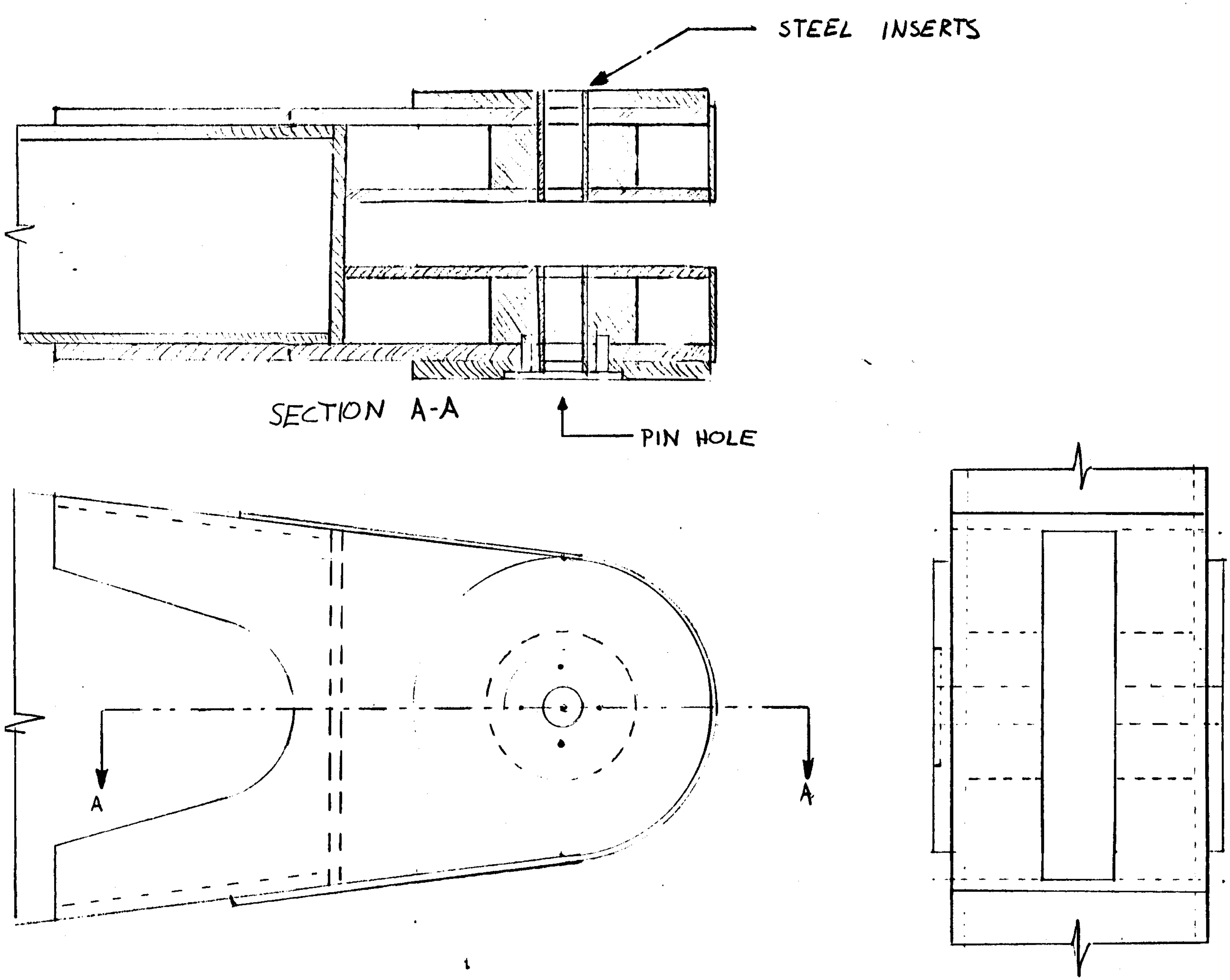
ROD EYE

ACTUAL SCALE

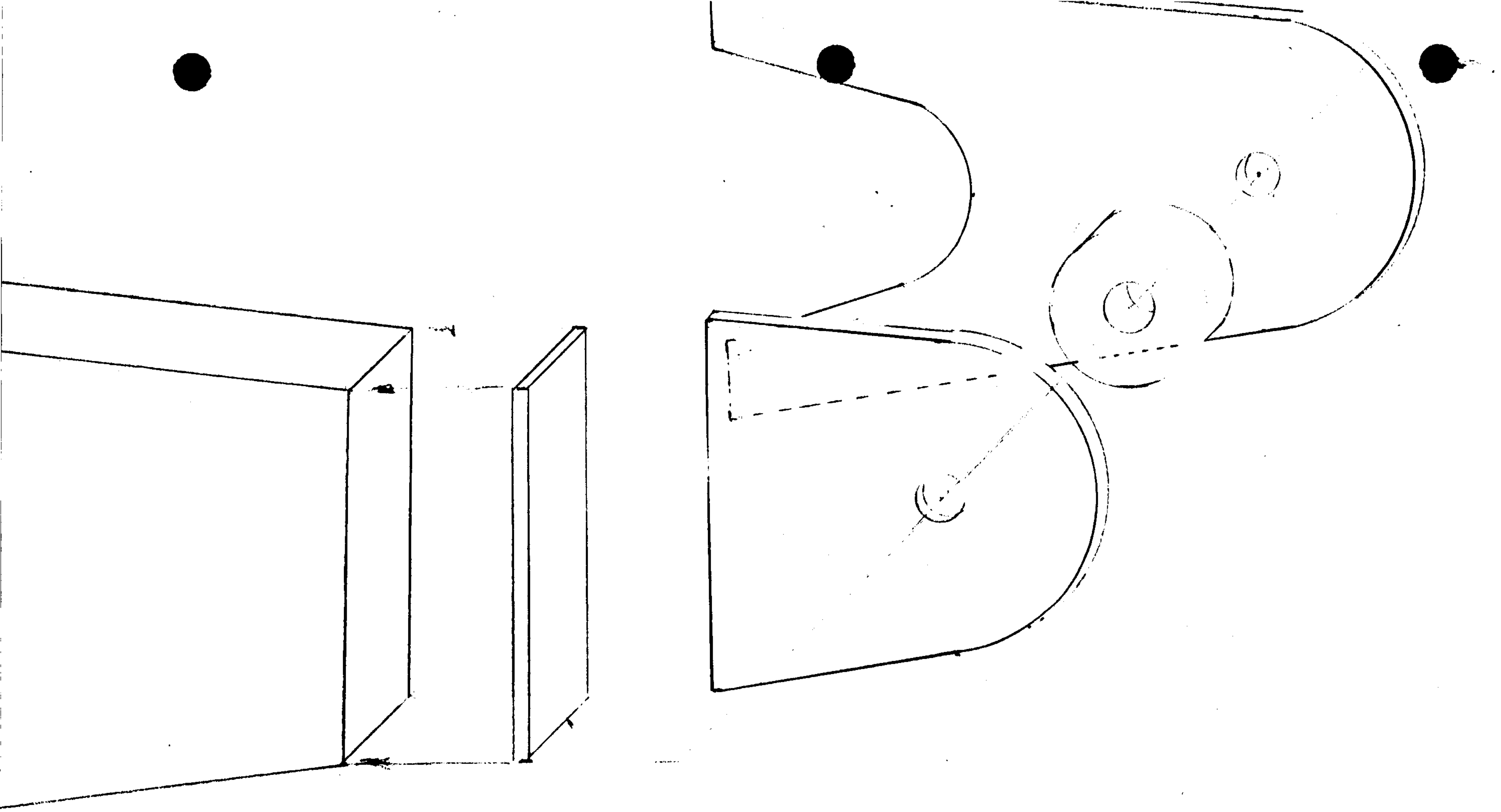


PIN / TYPICAL
1" = 2"

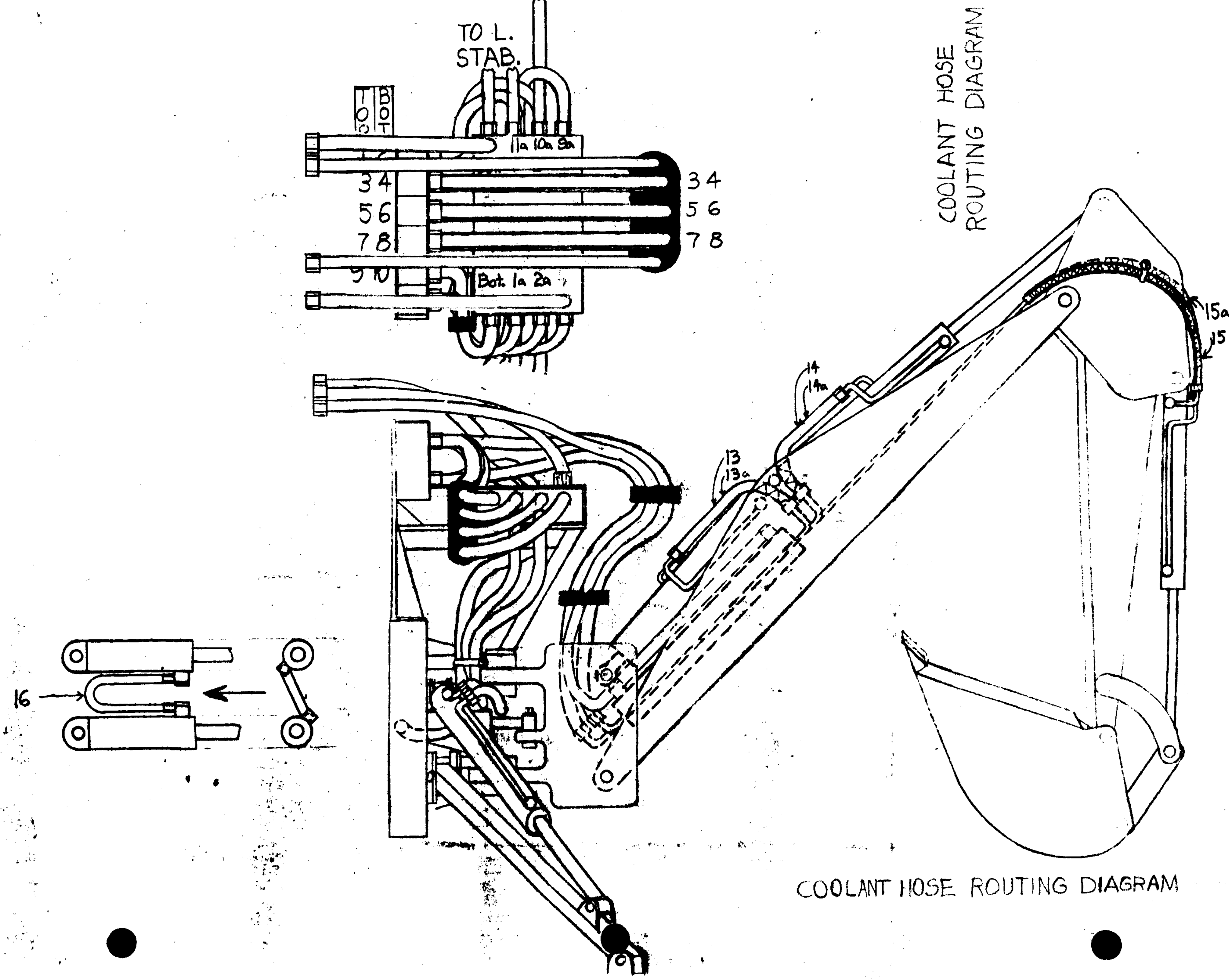
$\frac{1}{4}$ SC[illegible]E



PIN CONNECTION
(TYPICAL)



PIN CONNECTION
RECEIVING END
HALF SHOWN



COOLANT HOSE ROUTING DIAGRAM